AUTOMOBILE ENGINEER

DESIGN · PRODUCTION · MATERIALS

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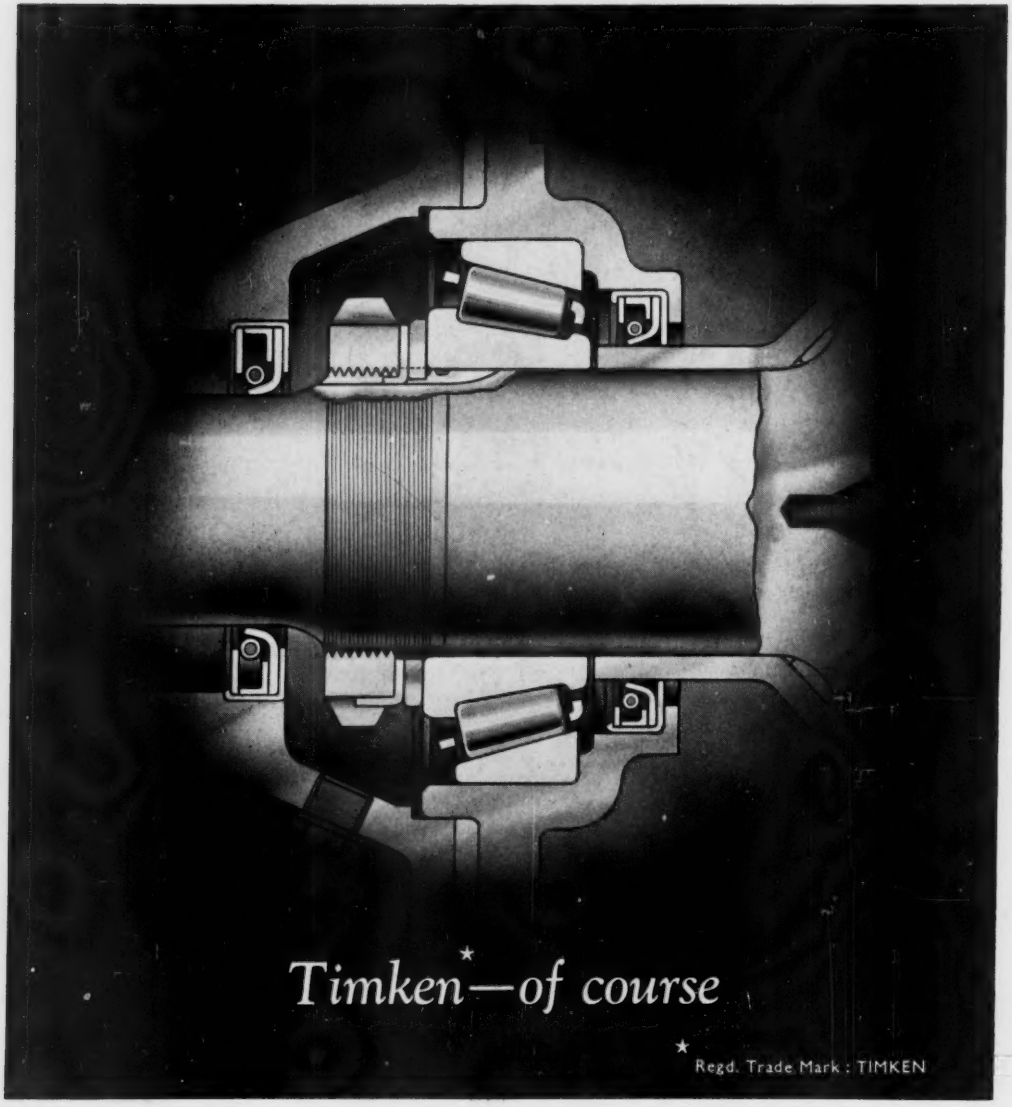

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ESC laminated and coil springs produced from our special quality alloy steels are supplied in large quantities to the motor trade.

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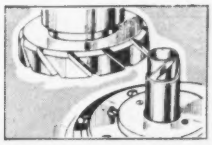

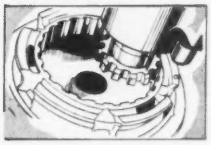

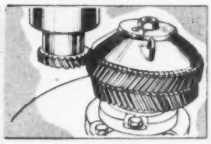

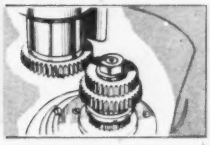

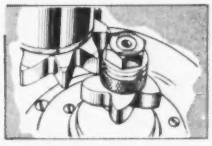

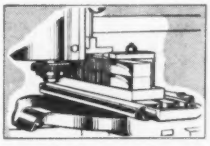

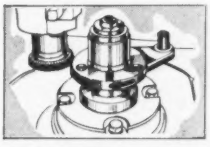

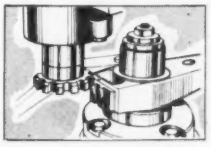

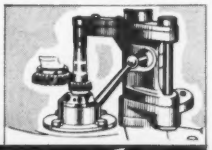

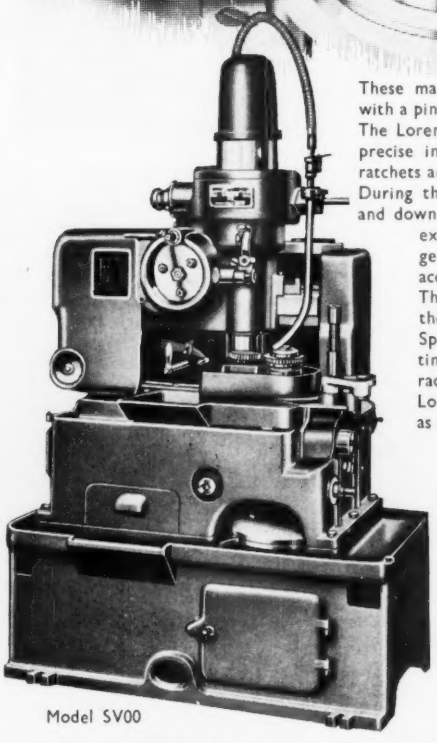

Model SV00

These machines shape by the generating process with a pinion type cutter for spur and helical gears. The Lorenz method is ideal for the production of precise internal and external gearing, sprockets, ratchets and other profiles.

During the cutting stroke the tool is moved up and down and rotated so that the tool and work execute the movement of two meshing gears. This method ensures maximum accuracy and smooth flanks of the work. The tool is re-sharpened by simply grinding the tooth face.

Special attachments are available for generating helical gears, gear trains, face gears, rack gears, shaping between centres, etc. Lorenz gear shapers are made in five sizes as shown below.

AUTOMATIC GEAR SHAPERS

MODEL	S 00	SV 00	S 5	S7.500	S7.1000
Pitch diam.	7¼"	7¼"	17¾"	19¾"	40"
Face width	2"	2"	3"	5¼"	6.5/16"
Diam. pitch/ module	6/4	6/4	5/5	3.6/7	3.6/7

AUTOMATIC GEAR HOBBERS

High production machines of extreme accuracy for the manufacture of spur, helical, worm wheels, and special gearings are made for work of pitch diameters from 23⅞" to 63" and diam. pitch/module 4.23/6 to 1.59/16.

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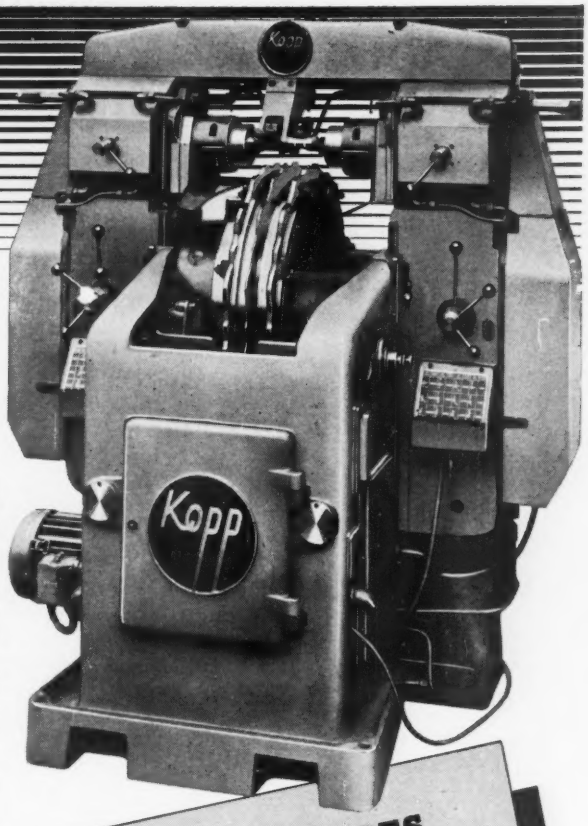
For fast production of contours to gauge limits



These high production machines are designed for the milling of simple or intricate profiled components, and are particularly suitable for the production of large quantities of a single design.

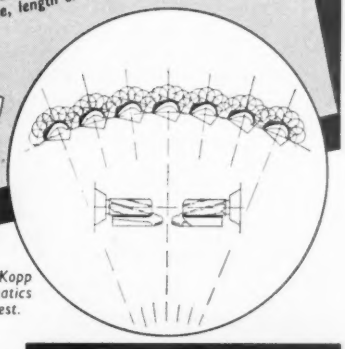
A continuously revolving drum forms a base upon which are mounted individual fixtures for holding the workpieces. Spindle heads to the right and left of the drum carry simple end milling cutters, and as the table revolves the tool approaches and tool feeds are controlled automatically by a hydraulic system. Automatic quick travel is provided between actual milling operations. Resetting of the workpiece is unnecessary, the cutters automatically following the most involved curvatures, the operator having only to load and unload the components.

EXAMPLE

TURBINE BLADES

Outer contour milled in 45 seconds with two simple milling cutters

Cutter diameters 20 mm. Cutter revolutions 250 per minute, length of cut 25 mm.



A catalogue describing Kopp profile milling automatics is available on request.

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187 F 37

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Photo. Lambert, Bury St. Edmonds

then double it!

"We should normally expect to draw pistons around 50,000 miles—but we found that with Shell Rotella Oils in use we go upwards of 100,000 miles," say Mulley's Motorways, of Ixworth and Bury St. Edmunds, Suffolk, giving these examples:—

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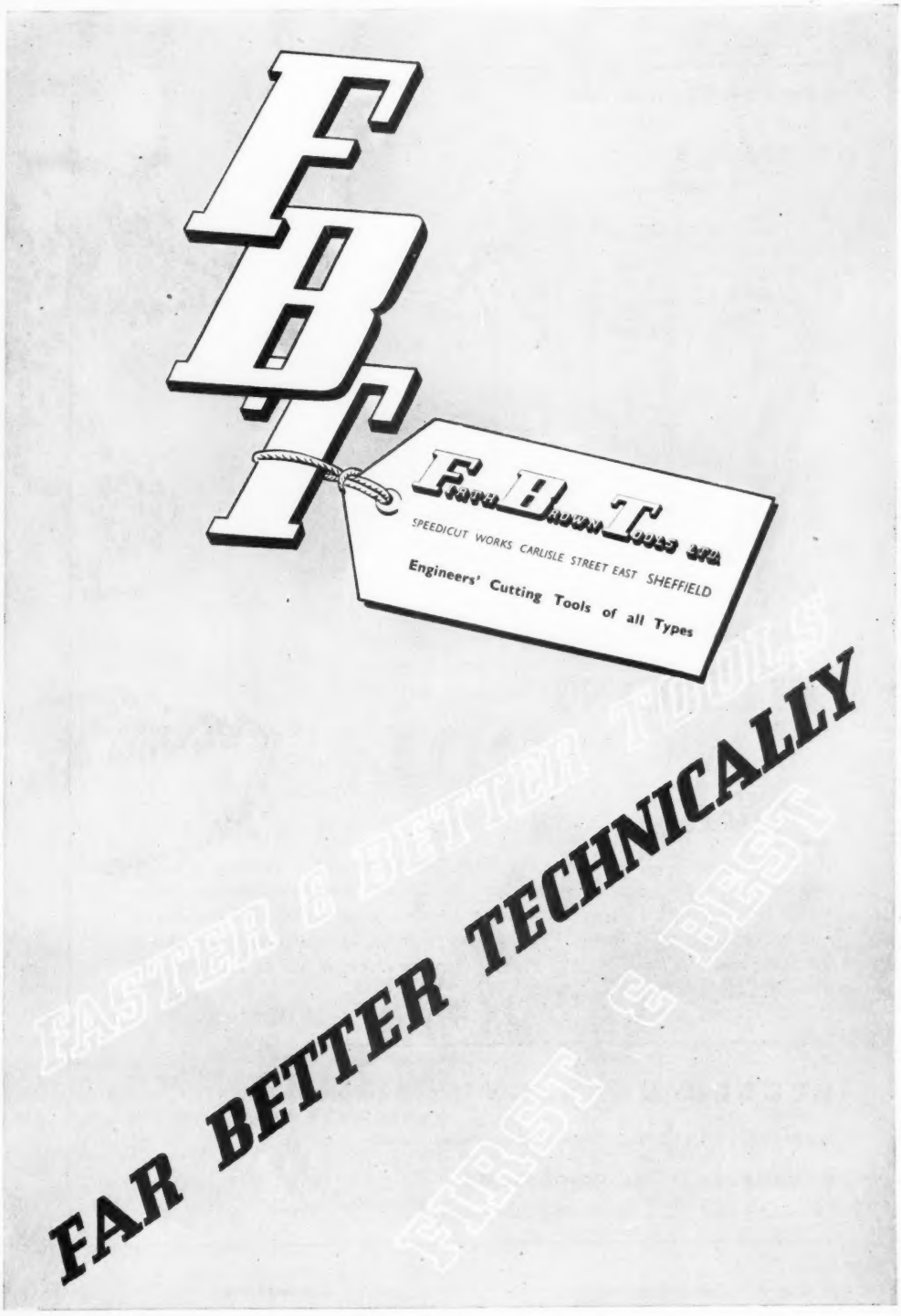
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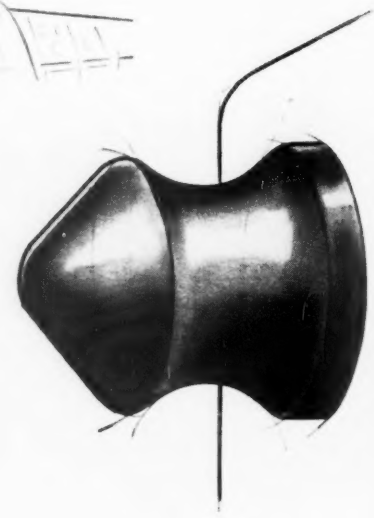

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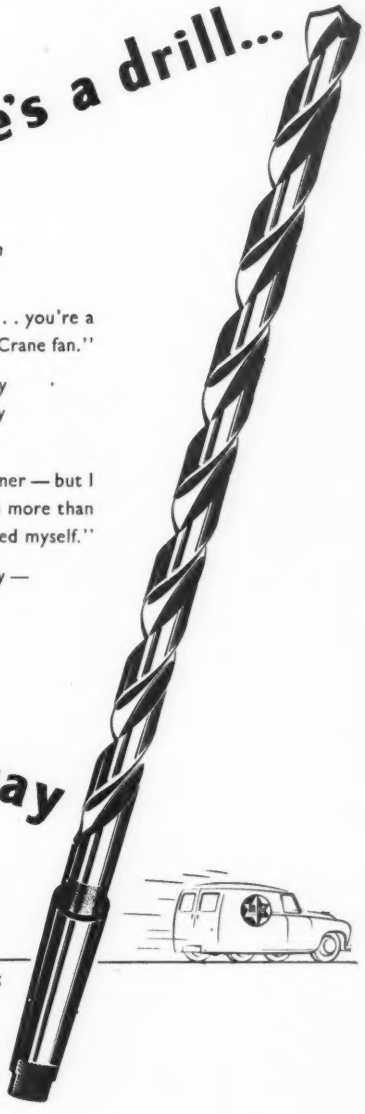

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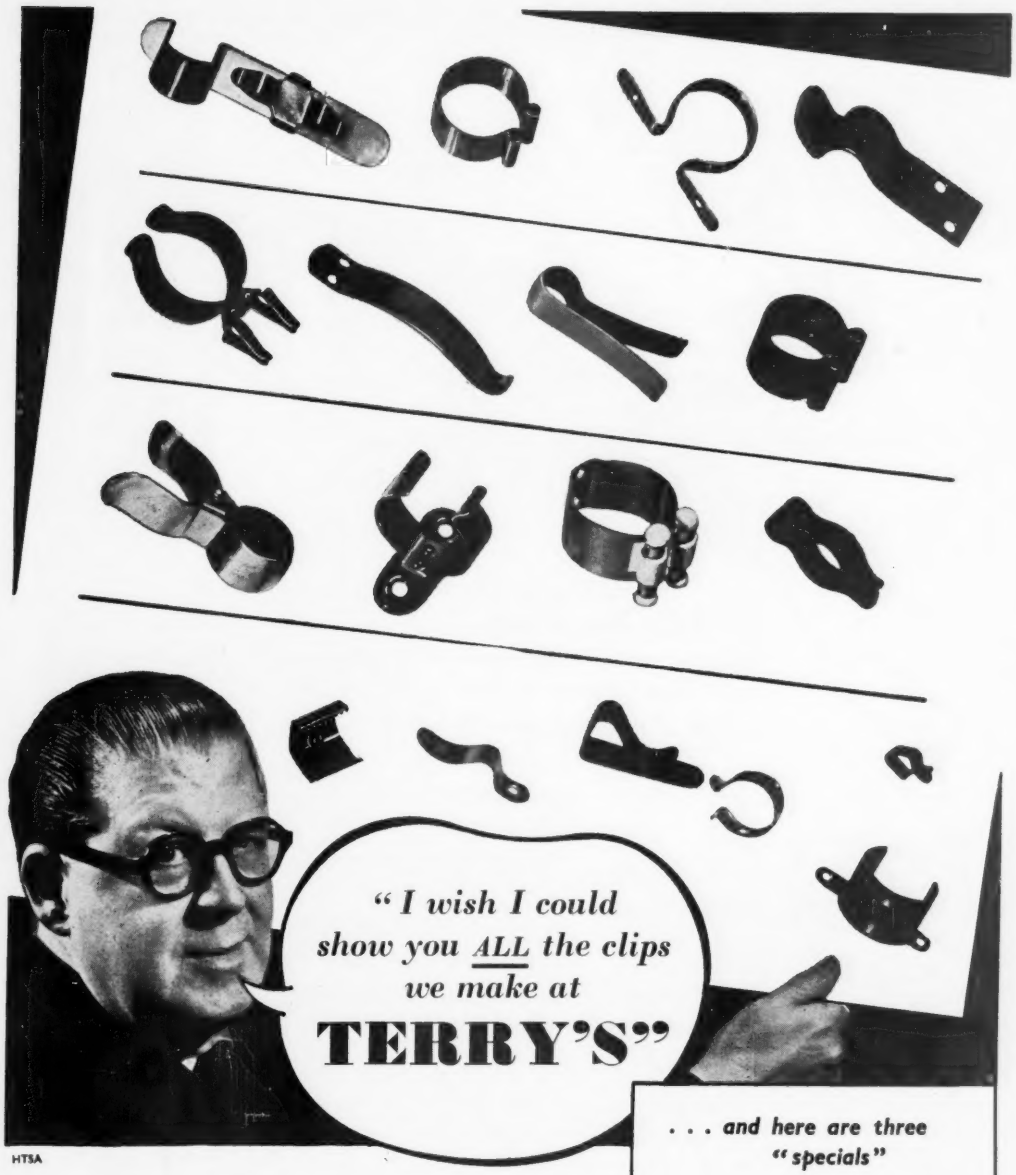

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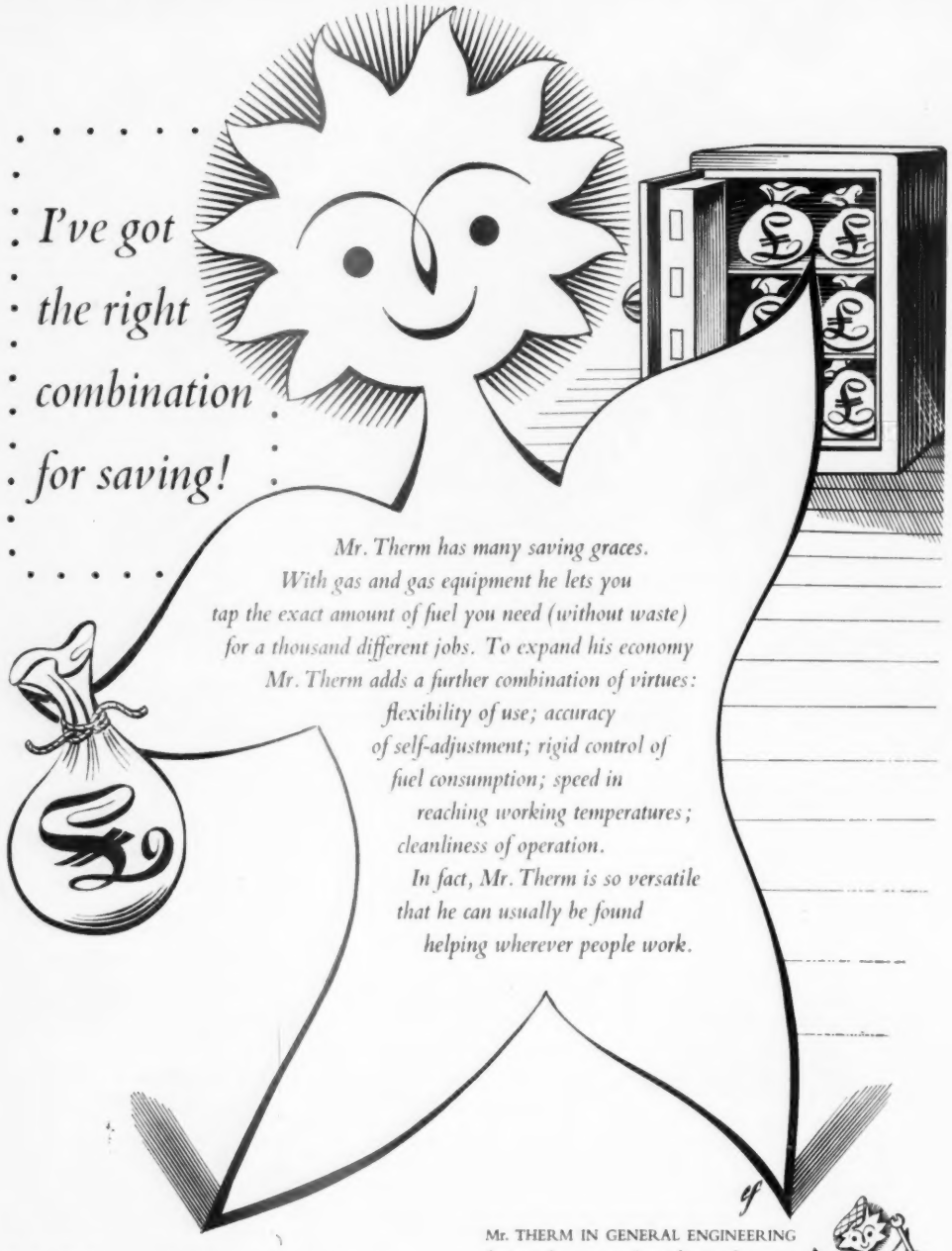

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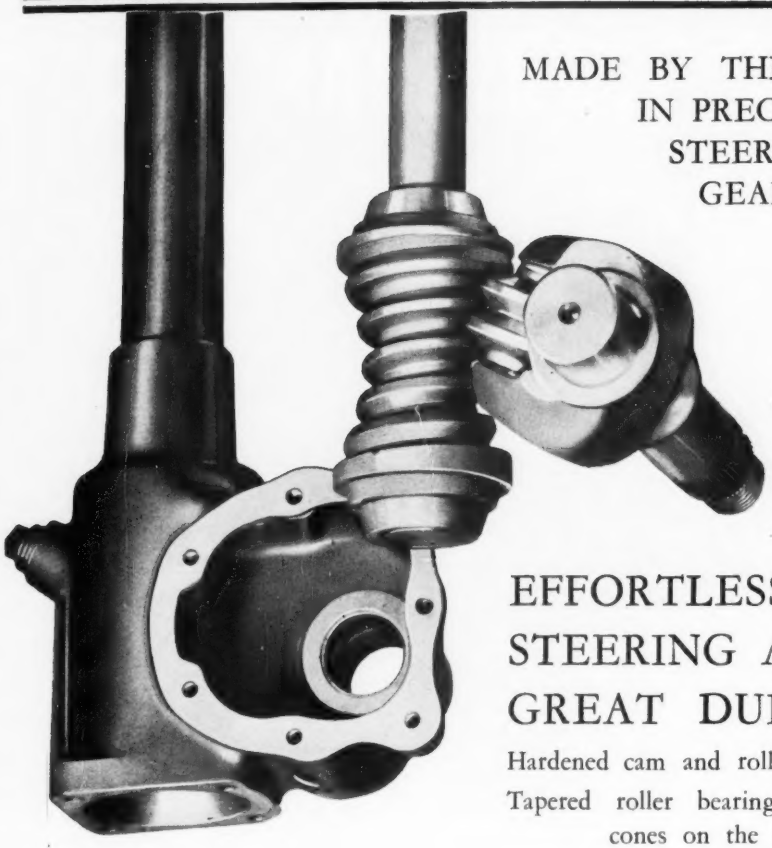

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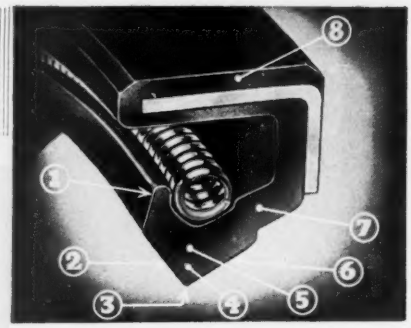

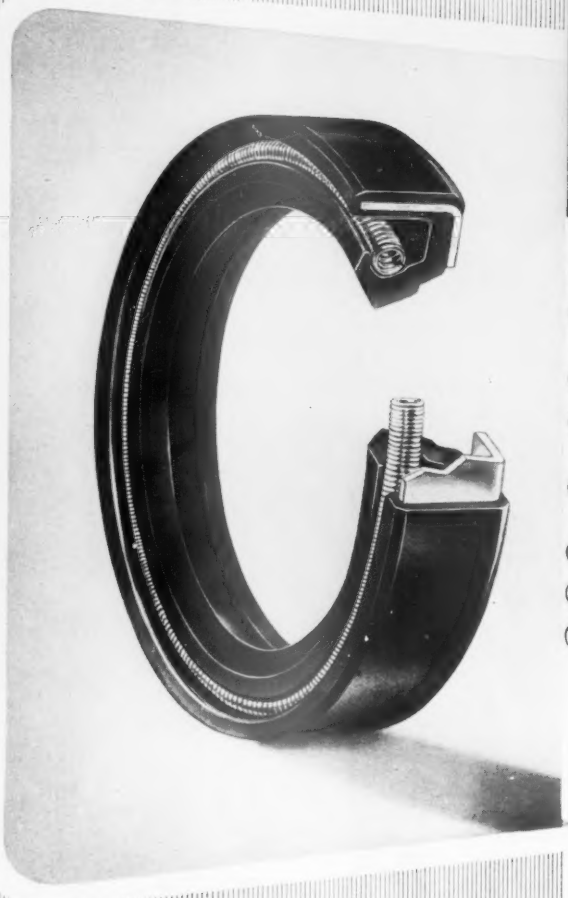

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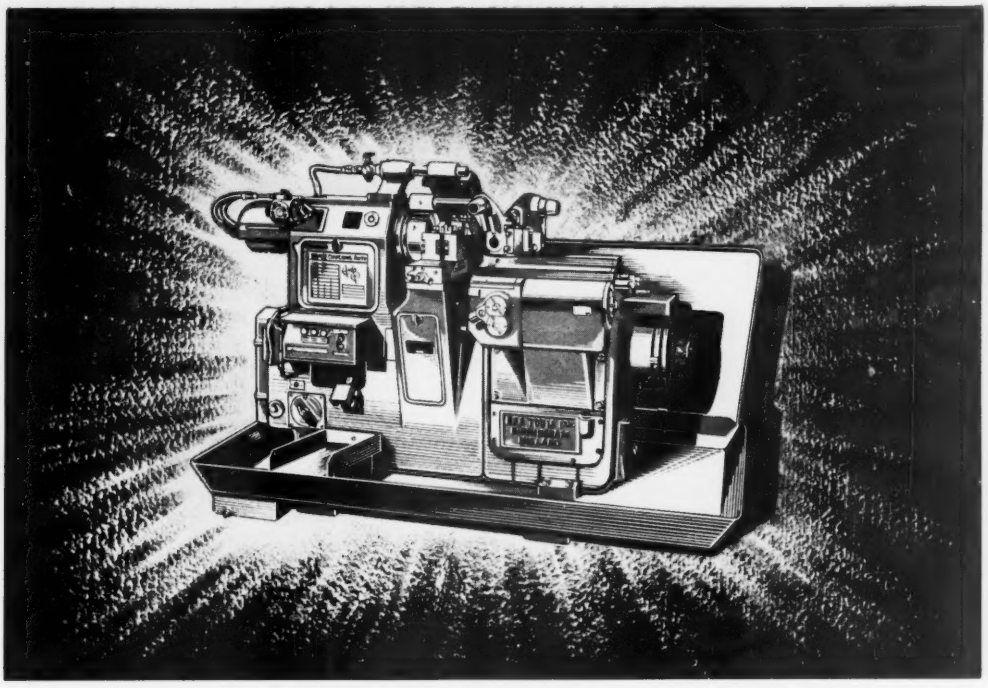

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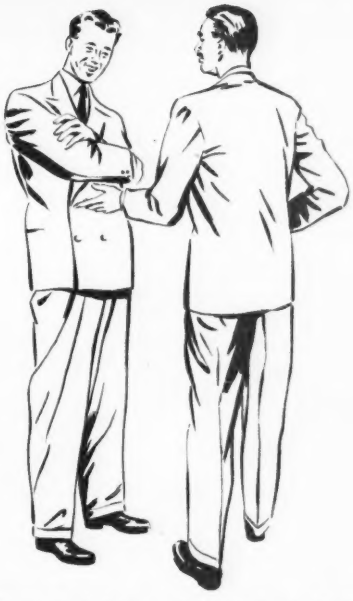

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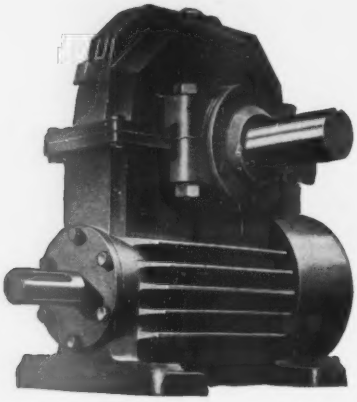
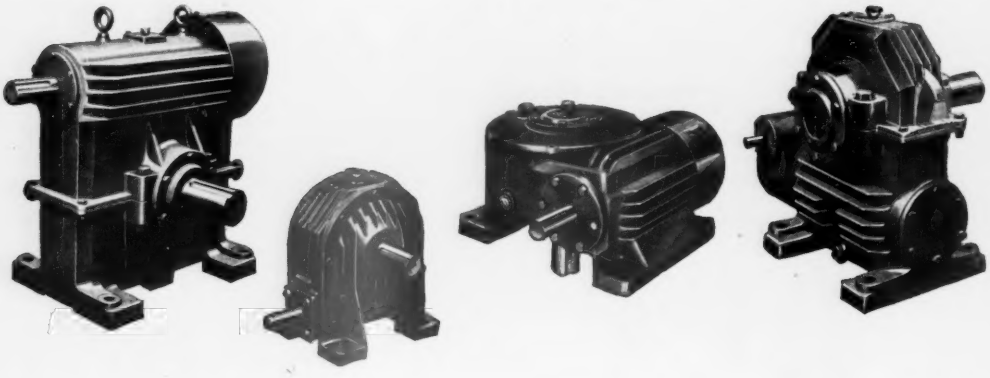
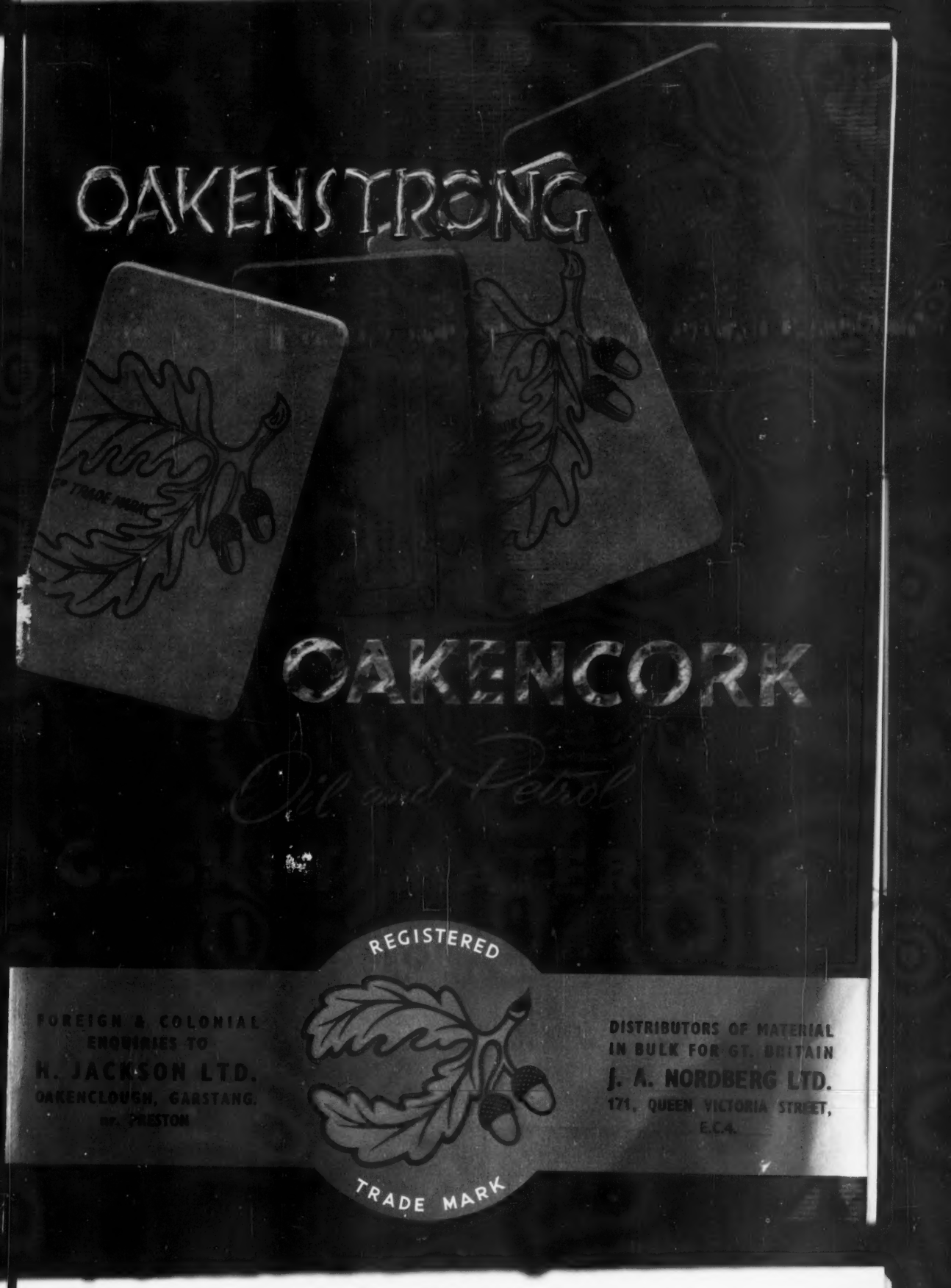


meet the boys . . . Some of our customers tell us they don't know how many types of Totally Enclosed Worm Reduction Units we make. Well! Here they all are. The big fellow at the top is our Underdriven (Standard) Worm Reduction Unit—comes in any size from 3″ to 24″ centres and ratios from 5/1 to 70/1. Then, first on the left below, our Overdriven W.R.U. — same range of centres and ratios but with driving shaft at top. Little Tich is our F. Type for fractional horsepower — 3 sizes: 1½″, 1¾″, and 2½″ centres each with 3 different relative positions of the input and output shafts. It may be mounted with the feet in any plane. The fat flat chap is our Vertical Type—same range as the Standard, but with a vertical wheel shaft that can be extended above or below the box. And last but far from least, our Double Reduction Unit for ratios up to 3000/1. Tough looking lot, aren't they! Tough, hard-working and reliable as a bevy of iron-jawed bank managers.

Holroyd Gear Reduction Units



JOHN HOLROYD AND COMPANY LIMITED · MILNROW · LANCASHIRE
CBC145


OAKENSTRONG
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J. A. NORDBERG LTD.
171, QUEEN VICTORIA STREET,
E.C.4.

Detachable oil filter

element easily inspected and replaced

This new AC partial-flow oil filter with replaceable element meets the increasing demand for a filter that can be removed for inspection and replaced every 8 10,000 miles. It provides a permanent installation for coupling into a by-pass oil supply system and the filtering element can be replaced without disturbing pipe connections. Sump oil is filtered on an average of ten times an hour. This means less motor wear and longer motor life — the objective of every motor manufacturer.

AC

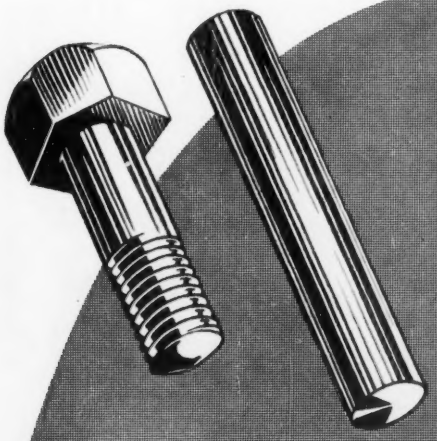
AIR CLEANERS · AIR SILENCERS · CRANKCASE BREATHERS
CAR HEATERS · FUEL PUMPS · SPEEDOMETERS · GAUGES
INSTRUMENT PANELS · OIL FILTERS · THERMOSTATS
SPARK PLUGS · WINDSCREEN WIPERS · DIE CASTING

If you are contemplating an important new design, or a modification of an existing one, take advantage of the facilities of the AC Technical Bureau. We probably can save you hours of planning and experimentation. Write to the AC Technical Bureau, 54, The Butts, Coventry; or telephone Coventry 61747.

AC-DELCO DIVISION OF GENERAL MOTORS LTD., DUNSTABLE, ENGLAND
and Southampton, Hants.

THE MACHINE TOOL APPROACH TO

HEATING FOR FORGING & UPSETTING



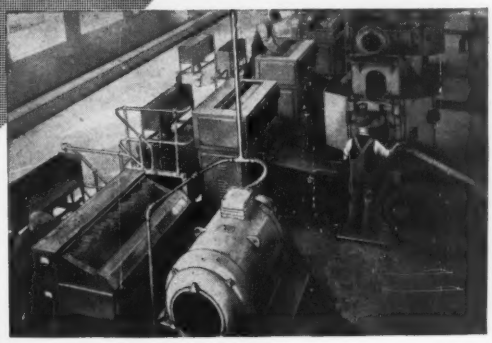
product $\frac{3}{4}$" black bolts.

process heating one end of pin for heading.

output 1,400 pins per hour.

work cost 8s. 6d. per 1,000 pieces.

equipment Birlec 80 kW automatic induction heater.



There are now 10 Birlec units in this black bolt shop.

Three station, hand-fed, unit for heating bars.

Standard Birlec induction heaters are available for heating bolt pins (from $\frac{1}{2}$" to 1" diam.) for heading. Automatic in operation, these machines deliver 1,400 pins per hour to the forgeman. Other designs of Birlec H.F. heaters help the forging industry to obtain "machine tool" qualities of speed, economy, cleanliness and regularity of output, over a wide range of bars, billets and other regularly shaped stock. Publication No. 85 deals with this subject; may we send a copy?

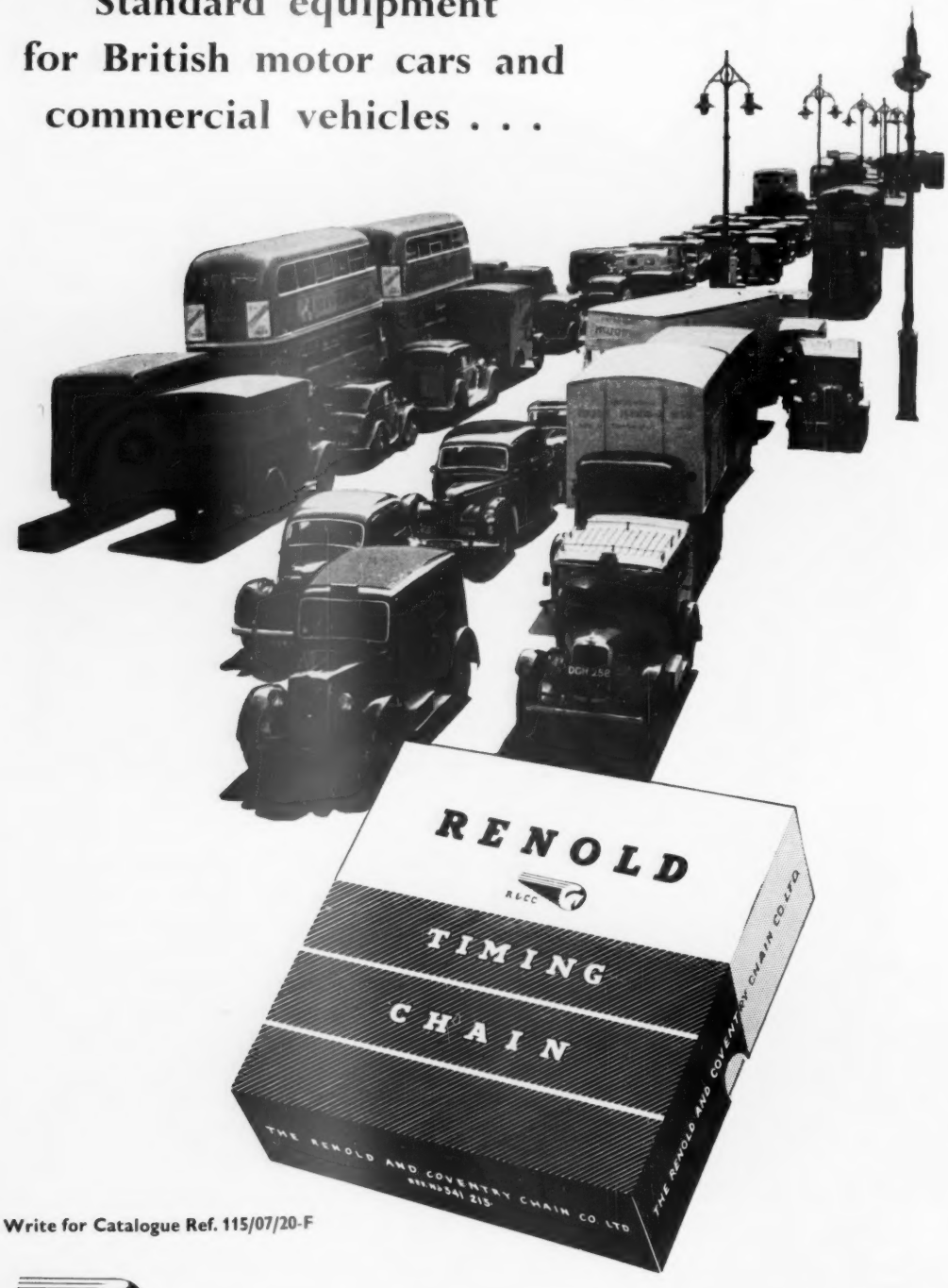
smb.6683

INDUCTION HEATING DIVISION

BIRLEC LIMITED

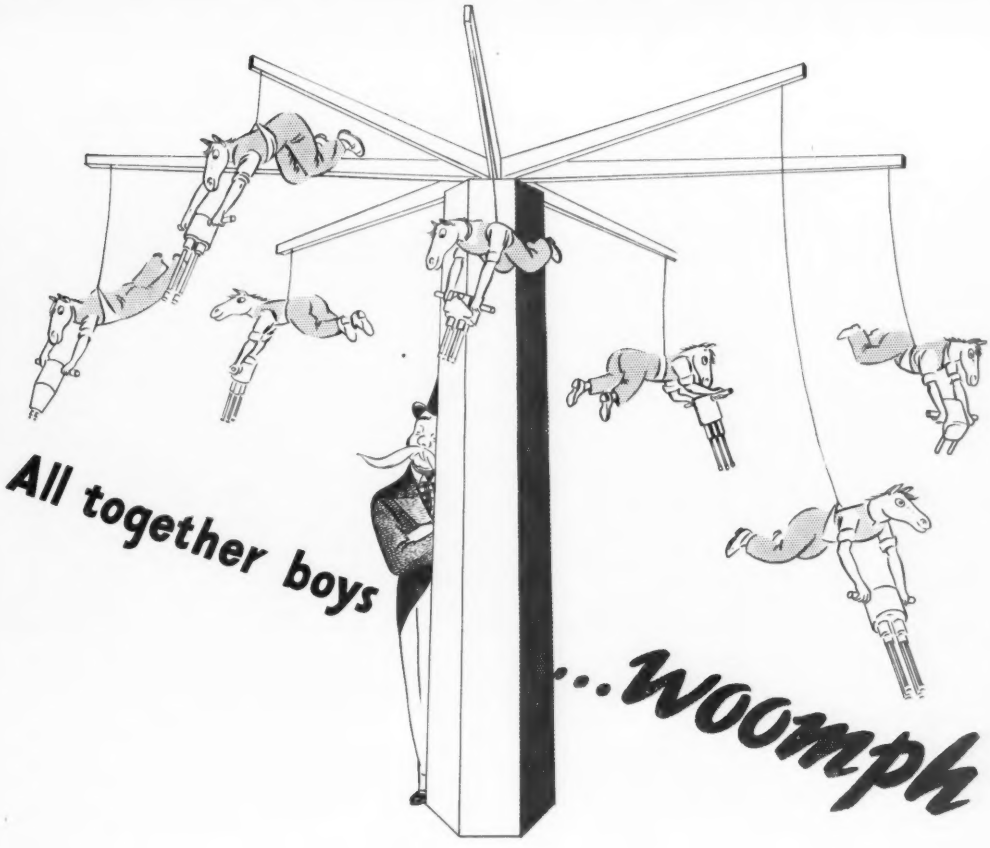
ERDINGTON · BIRMINGHAM · 24

Standard equipment for British motor cars and commercial vehicles . . .



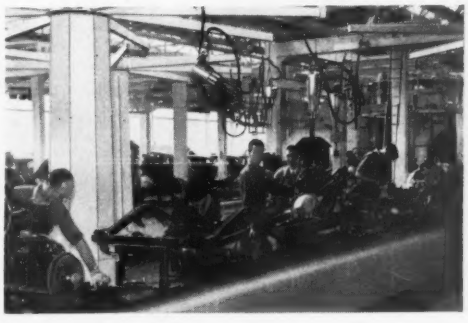
Write for Catalogue Ref. 115/07/20-F

R.C.C.C. THE RENOLD & COVENTRY CHAIN CO LTD · MANCHESTER



OUR LITTLE HORSES think it's a new and lovely game. Round they swing and one after another they are pulled down, each to do some assembly job on an Austin car. Little Joe will fasten on the engine. Frederick, Sebastian and Montmorency come down as a team and deal with the brake link. The Quads fasten the four wheel nuts in one operation.

All assembly in this new Austin Plant is done by our Little Horses and very proud of them we are. And a nice man with curly hair on the Austin Assembly Line said *he* was proud of them too.

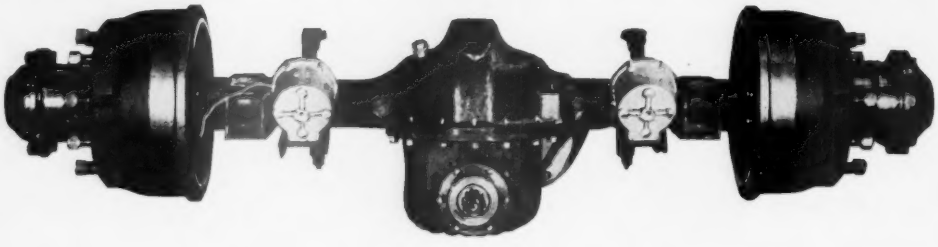


Desoutter

POWER TOOLS INCREASE PRODUCTION

DESOUTTER BROS. LTD., THE HYDE, HENDON, LONDON, N.W.9 TELEPHONE : COLINDALE 6346 (5 LINES) TELEGRAMS : DESPNUCO, HYDE, LONDON CRC237

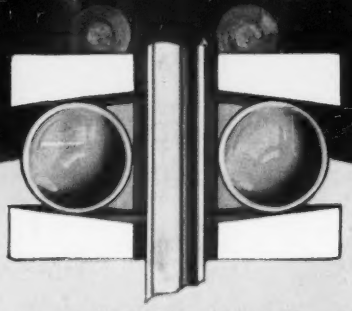
KIRKSTALL AXLES



KIRKSTALL FORGE ENGINEERING LTD
LEEDS

Telephone : Horsforth 2821

ACCURATE SPEED CONTROL



Iso-Speedic governors embody flybobs which consist of hardened steel balls rolling on ground tracks, eliminating all static friction, and resulting in a high degree of accuracy.

They are used on diesel engines and petrol engines, and are available for other speed control applications.

Iso-Speedic governors can be supplied which give control within 0.3% and are used on generators for radar and television. Other Iso-Speedic governors are available where a lower degree of accuracy is sufficient.

The services of our engineers are at your disposal.

Iso-Speedic

THE ISO-SPEEDIC COMPANY LTD., COVENTRY

The bodywork of the Austin A.40 Sports Model is fabricated from aluminium alloy sheet and strip supplied by T.I. Aluminium.

The radiator cowling is an extremely intricate pressing in T.I. alloy 111.

We are proud that the Austin Motor Company choose to

PUT Aluminium FIRST
AND TI IN FRONT OF IT

T.I. ALUMINIUM LTD

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ALUMINIUM AND ALUMINIUM ALLOY INGOT, SLABS, BILLETS, SHEET, STRIP, TUBES AND EXTRUSIONS TO ALL COMMERCIAL, A.I.D. AND LLOYD'S SPECIFICATIONS

Radiator Production

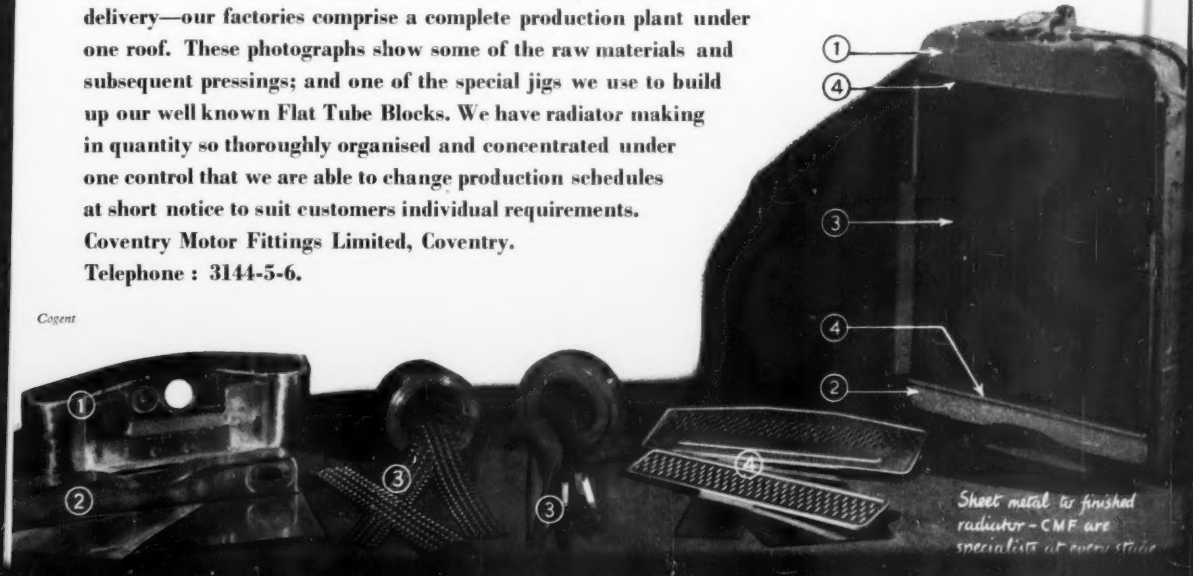
from A to Z

COVENTRY MOTOR FITTINGS

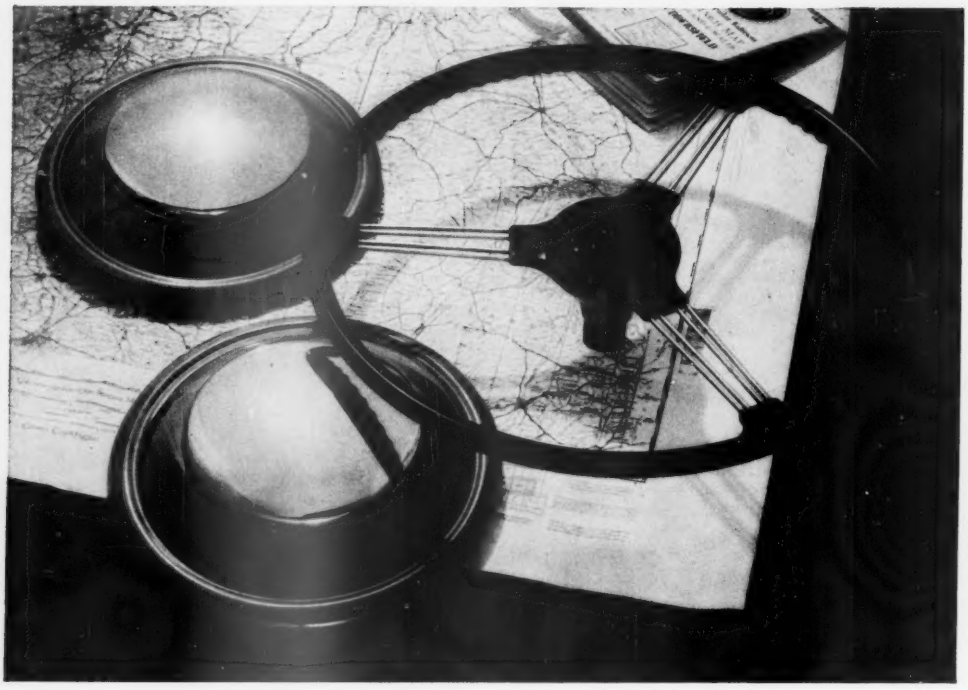
EST'D CMF 1902

Here at C.M.F. in Coventry we make complete radiators for most of the leading motor manufacturers. We do the whole job from design to delivery—our factories comprise a complete production plant under one roof. These photographs show some of the raw materials and subsequent pressings; and one of the special jigs we use to build up our well known Flat Tube Blocks. We have radiator making in quantity so thoroughly organised and concentrated under one control that we are able to change production schedules at short notice to suit customers individual requirements.
Coventry Motor Fittings Limited, Coventry.
Telephone : 3144-5-6.

Cogent



Sheet metal to finished radiator - CMF are



"SILVER FOX" . . . *the metal of the age*

Registered Trade Marks

STAINLESS STEEL

Motor vehicle components and fittings made of Stainless Steel cause no anxiety to the owner—the surface can never wear or peel off; the metal is stainless right through. There is no need to polish; just a wipe with soapy water and they come up bright and unspotted for the whole life of the car.

When the nation's needs permit, "Silver Fox" will be available to the discriminating motorist in ever-increasing quantities.

Our colour photograph shows Hub caps by courtesy of the Ford Motor Company. Steering Wheel by courtesy of Wilmot Breeden Limited.

THE UNITED STEEL COMPANIES LTD

SAMUEL FOX & COMPANY LIMITED

Associated with The United Steel Companies Limited

STOCKSBRIDGE WORKS · Nr. SHEFFIELD · ENGLAND

F259

HERBERT

The Lubricar

A PORTABLE servicing unit carrying all the equipment and lubricants for oiling and greasing any machine, and a pump and filter for salvaging used oil from headstocks and gear boxes.

In charge of a suitably trained operator, working on a regular routine, the LUBRICAR ensures efficient lubrication; practically every machine in the plant can be completely serviced at regular intervals, preventing breakdowns and loss of production.

IMMEDIATE DELIVERY

Full details from Factored Division, Red Lane Works,
Phone: 89221 (10 lines)

ALFRED HERBERT LTD · COVENTRY

LOOK

where goods are loaded

or passengers are carried

...YOU SEE DURALUMIN

REGD. TRADE MARK

Look where you will in the world of transport and you will almost certainly see "Duralumin". It is only one-third the weight of steel and yet as strong. By its use many operating costs can be reduced, and carrying capacity increased. Petrol consumption is lower, and there is less wear on tyres. The decreased overall weight of the vehicle saves the cost of the road fund licence. As "Duralumin" does not rust and can be left unpainted, maintenance costs are much decreased.

The properties of "Duralumin" include a remarkable capacity for absorbing shocks. It is non-toxic, hygienic and easily cleaned—of importance where the carriage of foodstuffs is concerned—and it does not warp when wet.

Whilst shortages exist at present, this is a passing phase. Now is the time to consider the use of aluminium alloys in the design of vehicles of all descriptions, so that the plans may be complete when the needs of rearmament are satisfied.

Our Development Department will advise entirely without obligation.

JAMES BOOTH AND COMPANY LIMITED BOOTH ARGYLE STREET WORKS · BIRMINGHAM · 7

When you want

quick friendly *personal service ..

Bolts, Nuts, Rivets, Screws, Studs & Washers in all metals and threads • Pressure Gauges, Reducing and Relief Valves, Steam Traps, Cocks and Valves for every service • Non-Ferrous Metals in all sections • Ferrous and Non-Ferrous Tubes and Fittings • Contractors Tools and Equipment • Carpenters and Engineering Tools, Screwing Tackle • Stainless Steel Buckets Utensils, Fittings and Equipment

ring up **Walkers!**

NOW Official Stockists FISCHER BALL AND ROLLER BEARINGS

* *Personal service means that your call is handled immediately by someone who can not only take* all *your instructions, but is ready and technically able to offer practical, helpful advice.*

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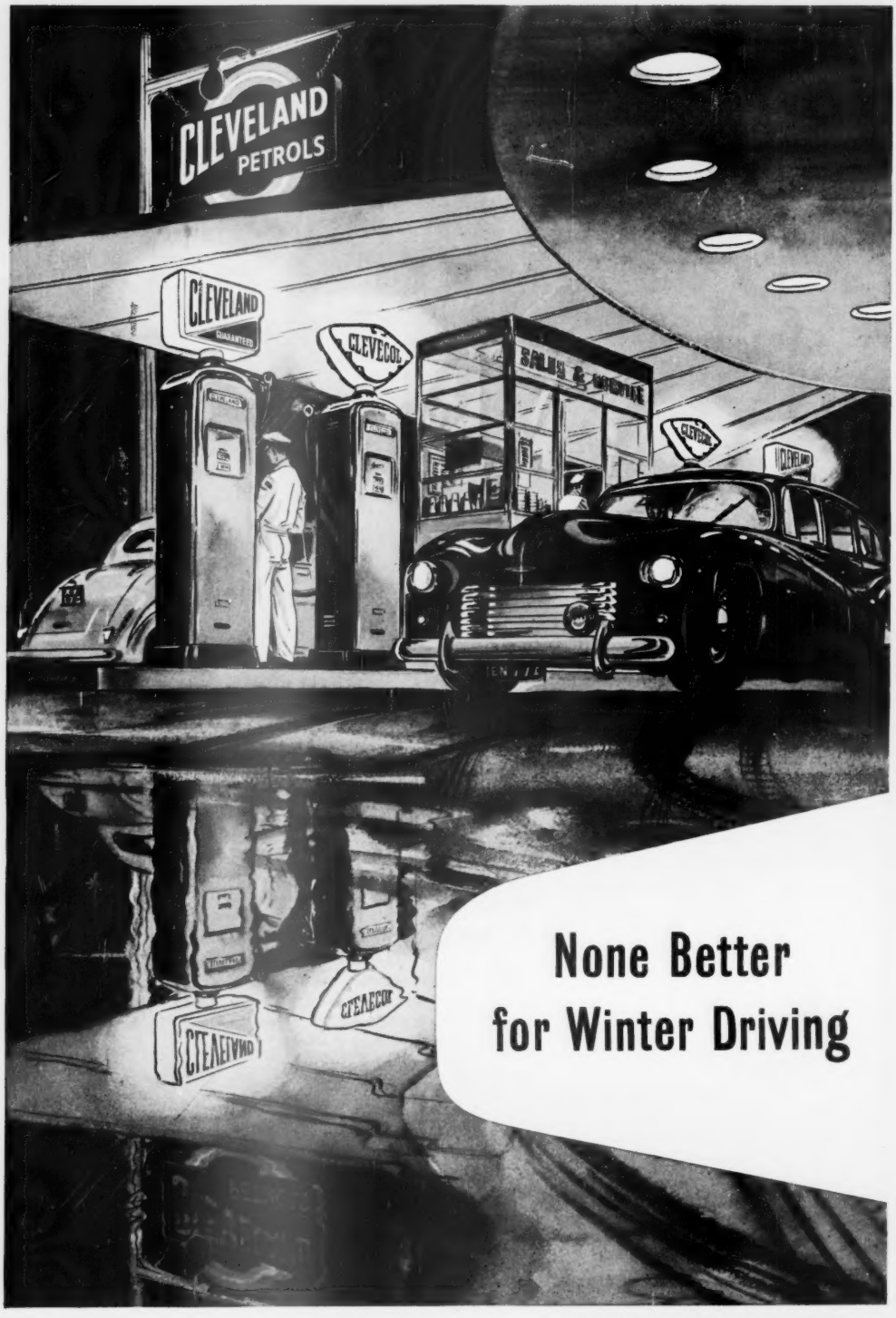
MIDDLE EAST BRANCH · BAHRAIN, PERSIAN GULF


CLEVELAND
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SALES & SERVICE
None Better
for Winter Driving

LEY'S

REQUIREMENTS,

Strength and Rigidity

Modern design calls for the finest material and foundry practice, with the result that more and more Ley's castings are being used for important parts.

Here is an axle housing for a famous firm, cast in Ley's 'Black Heart' malleable; with cored passages to suit modern lubrication practice:

LEY'S 'BLACK HEART' MALLEABLE CASTINGS:		LEY'S 'LEPAZ' MALLEABLE CASTINGS:	
Elongation:	18%	Elongation:	6%
Yield Point:	16 tons p.s.i.	Yield Point:	21 tons p.s.i.
Tensile Strength:	24 tons p.s.i.	Tensile Strength:	35 tons p.s.i.

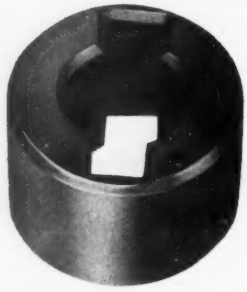
We offer your designers the full collaboration of our engineers.

LEY'S MALLEABLE CASTINGS CO. LTD., DERBY. TELEPHONE: DERBY 45671

EUROPE'S LARGEST MALLEABLE PRODUCERS

LEY'S

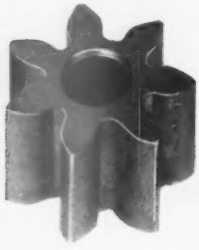
BOUND BROOK *POWDER METALLURGY*

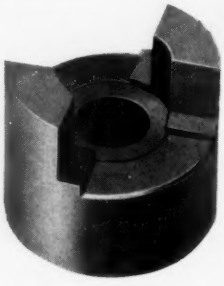


Knowing the ins and outs!

Powder metallurgy brings a new technique to the production of small structural and machine parts, often at lower cost with improved performance. Bound Brook sintered metal parts are the result of engineering knowledge, precision die making and specialised experience of compressed metal powders.

We shall be happy to advise you whether your parts can be produced in this way — whether the shapes required can be formed in dies permitting the necessary pressure to produce the correct density, etc. There are limitations, of course, but where powder metallurgy can be used it quickly justifies the change over from conventional methods. Send for leaflet "Sintered Components by Powder Metallurgy."





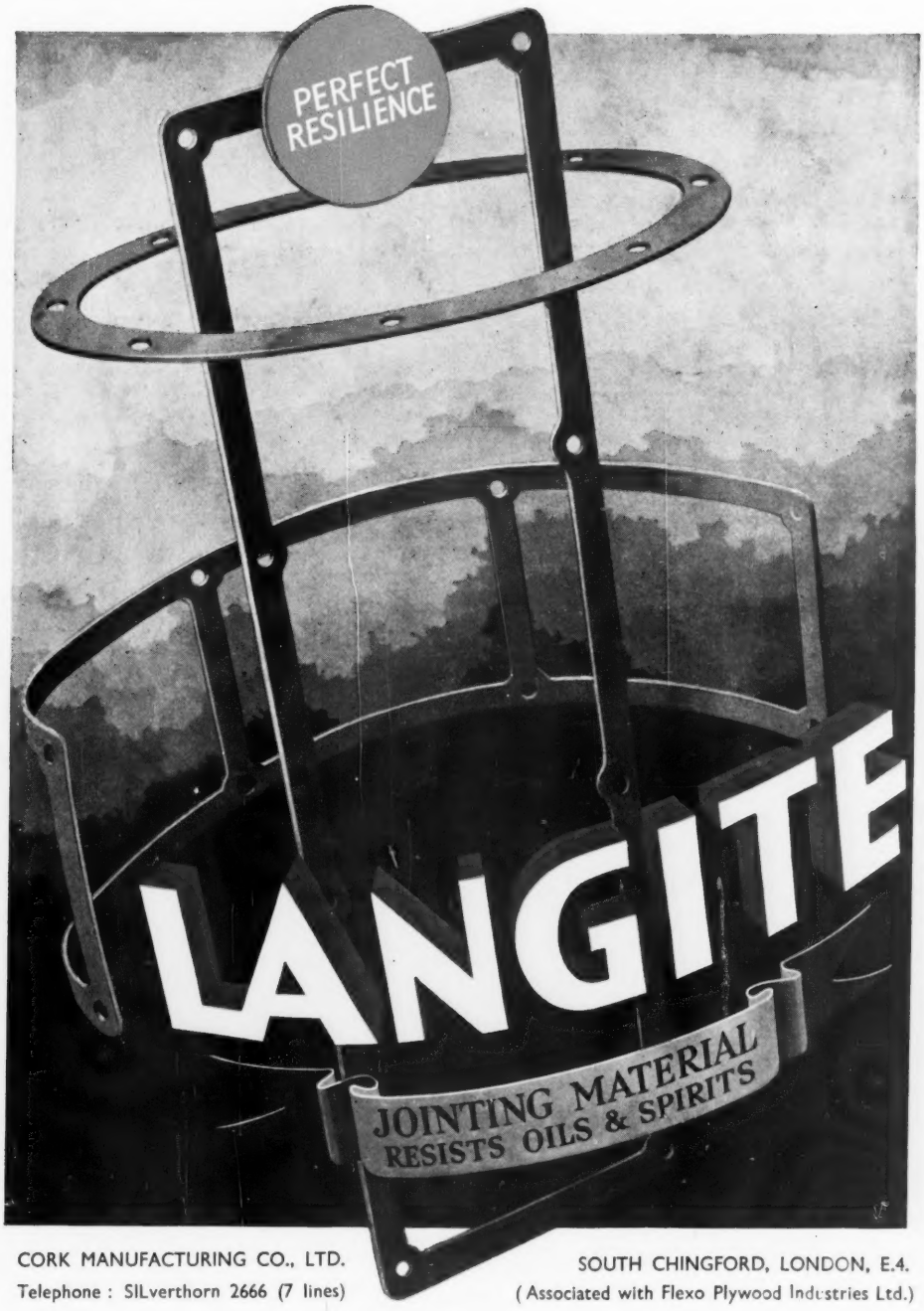
Powdiron

SINTERED METAL PARTS

BOUND BROOK BEARINGS LIMITED

BOUND BROOK BEARINGS LTD., TRENT VALLEY TRADING ESTATE, LICHFIELD, STAFFS

(A Birfield Company) *Telephone: Lichfield 2027 - 2028*



CORK MANUFACTURING CO., LTD.
Telephone : SILverthorn 2666 (7 lines)

SOUTH CHINGFORD, LONDON, E.4.
(Associated with Flexo Plywood Industries Ltd.)

*CARMET

COPPER CARBON

an eye to the future

rings
sleeves
bearings
seals

. . . . engineers with foresight have been quick to appreciate the advantages of CARMET*, and everywhere, more and more, they are specifying this material. Combining the strength of metal with the self-lubricating properties of Carbon; there is extremely low wear rate, a favourable co-efficient of friction and good resistance to high temperatures.

The MORGAN CRUCIBLE COMPANY LIMITED

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Telephone: Battersea 6822. Telegrams: "Crucible, Sowphone, London."

Our Engineers will be pleased to advise you on its application.

* *CARMET is a Registered Trade Mark of The Morgan Crucible Company Ltd.*

C35

A clean sweep

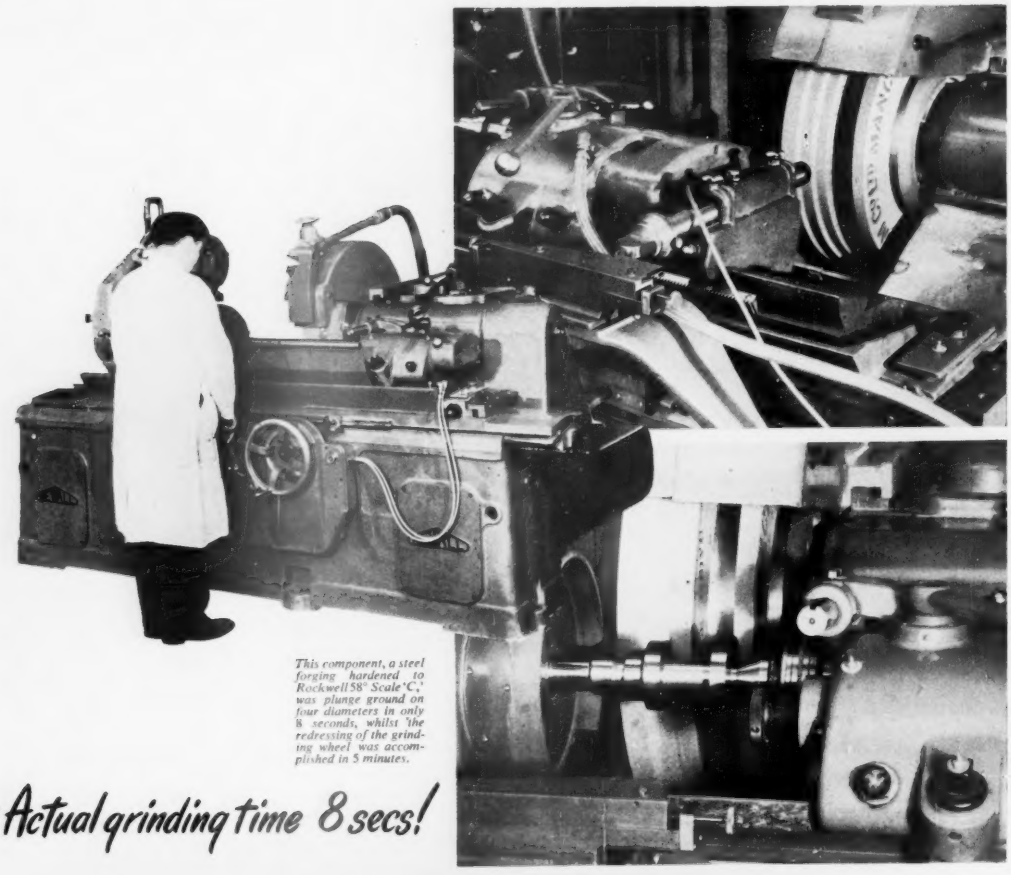
Taking up dust wherever it goes, the Hoover "Dustette" does its job, efficiently and hygienically. In workmanship and materials it reflects the highest quality, down to the very screws. And here's where we take a bow. Because they measure up to Hoover-high standards Linread screws *are* used in all Hoover products. Why Linread — because intensive metallurgical research and strict control in manufacture ensure that our screws are as good as screws can be. Hoover is only one of many household names with whose products we are proudly associated. Like the buyer of a Hoover cleaner, these manufacturers are discriminating in their choice. They will have nothing but the best, and for all types of small fasteners that is synonymous with Linread.

Linread LIMITED

We supply all industries requiring small fasteners of the highest quality to hold together the products they manufacture. Our specialists are always at your service to advise and assist you with your own particular problems.

LINREAD LTD. STERLING WORKS COX ST. BIRMINGHAM 3

Beardmore

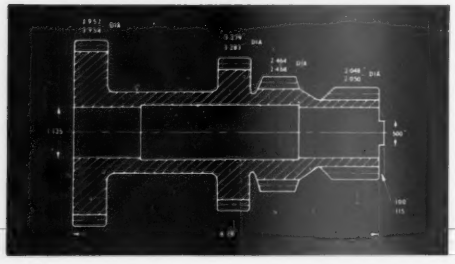


This component, a steel forging hardened to Rockwell 58° Scale 'C,' was plunge ground on four diameters in only 8 seconds, whilst the redressing of the grinding wheel was accomplished in 5 minutes.

☆ Actual grinding time 8 secs!

Grind all diameters at one plunge..

In common with other leading automobile manufacturers, the Ford Motor Co., (Canada) Limited employ only the most modern production methods. Amongst their tools is this Newall 10″×24″ L.A. Grinding Machine. With this machine, all diameters of Gear Transmission Countershafts can be ground at one plunge—the actual grinding time taken being only 8 seconds. The L.A. Grinding Machine has proved to be the right tool for high and economic outputs.



◀ gear transmission countershaft

P1295

. . . here's another Die Casting Problem

Good clean finish and pressure tightness are essential for the filter heads of these well-known Oil Filters. John Dale Aluminium Alloy Gravity Die Castings have achieved these results.

Consult **JOHN DALE** *Limited*
about Aluminium Alloy Gravity Die Castings

JOHN DALE LTD. (DEPT. TS.3), LONDON COLNEY, ST. ALBANS, HERTS. **Tel. : Bowmansgreen 2266**

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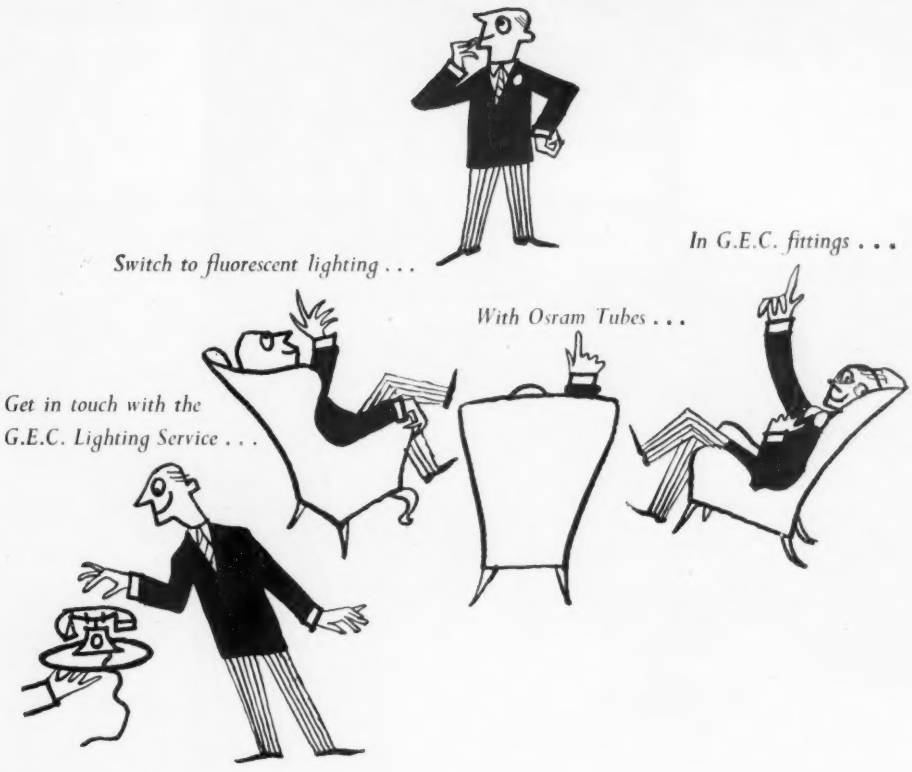
DUALLOYS
BAINBRIDGE
PLAIN BEARINGS
DUALLOYS LIMITED · BODEN WORKS · CHARD · SOMERSET

DBI

Brainwaves?



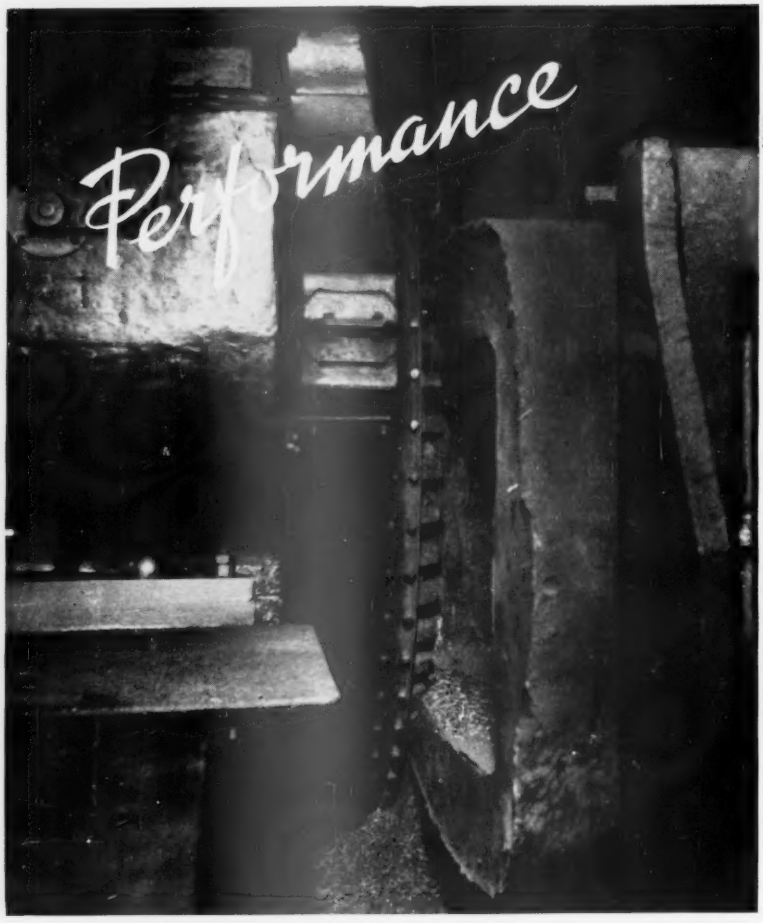
Fluorescent lighting?

Is your lighting really effective and economical? For instance, a modern fluorescent installation would cut your consumption of current by over half, for the same amount of light as you are now getting. Alternatively, you could have over double the light at no extra cost! The people to guide you are the G.E.C., pioneers of fluorescent lighting and the first to establish an Advisory Lighting Service with every technical help on call. Gentlemen, you'll save a lot of talk . . . if you see the G.E.C. first.

Osram

tubes in G.E.C. *fittings*

THE GENERAL ELECTRIC CO. LTD., MAGNET HOUSE, KINGSWAY, LONDON, W.C.2



Photograph by courtesy of the Distington Engineering Co. Ltd.

The above photograph shows a large rotary planing machine fitted with 60 Cutanit-tipped tools and making a heavy roughing cut across a 10 ton cast iron ingot mould.

The tools are of Cutanit Grade SW which is ideally suited for this class of work.

For Maximum Performance

Cut with Cutanit

TRADE MARK

CEMENTED CARBIDE

A METRO-CUTANIT PRODUCT

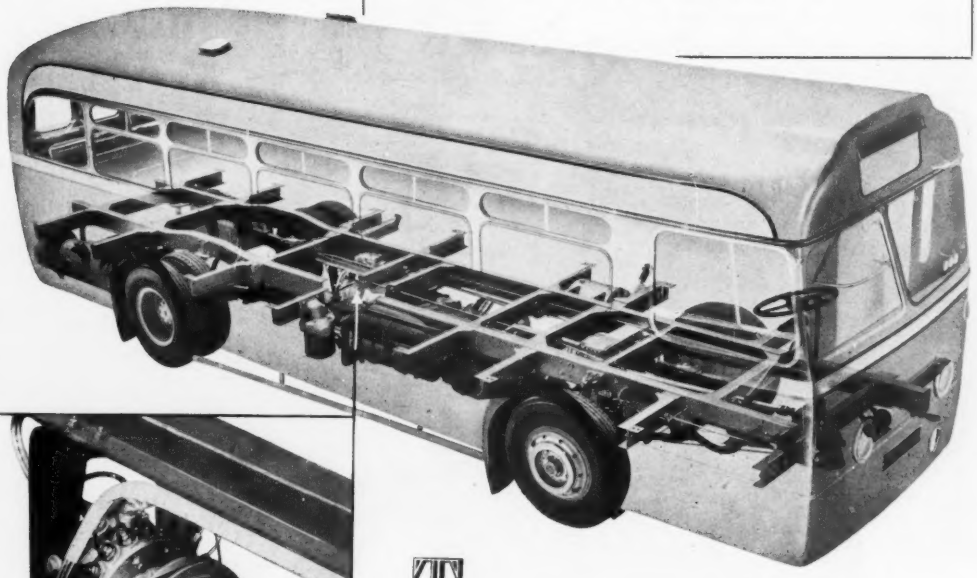
WM JESSOP & SONS LTD
BRIGHTSIDE WORKS SHEFFIELD

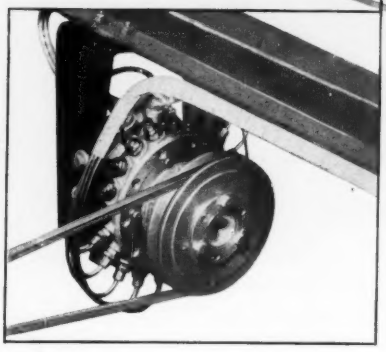
J J SAVILLE & CO LTD
TRIUMPH WORKS SHEFFIELD

THE SIGN OF QUALITY

the 'Inside story' of the *Syndromic*

AUTOMATIC MILEAGE CONTROLLED CHASSIS LUBRICATOR





THIS X-ray photograph of a coach is in itself unusual but the really interesting feature is the " Syndromic " chassis lubricator with which it is fitted.

This lubricator gives a predetermined and measured 'shot' of oil to each individual bearing every mile, operating efficiently and automatically all the time the vehicle is running. Once fitted and regulated it requires no attention other than keeping the tank replenished with oil.

Now see the advantages . . . no 'dead' time for periodical lubrication—more pay loads . . . wear practically eliminated on spring shackles, king pins, steering swivels, etc.—no replacement costs.

Why not fit your vehicle with " Syndromic " automatic mileage controlled chassis lubrication?

Let us send you full particulars of the system.

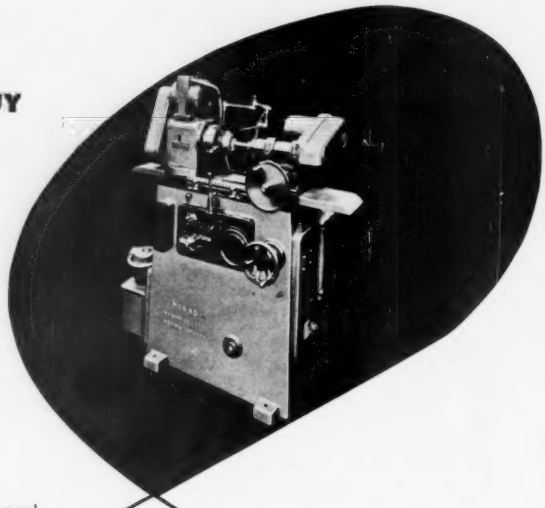
TECALEMIT

The Authority on Lubrication

PLYMOUTH. ENGLAND

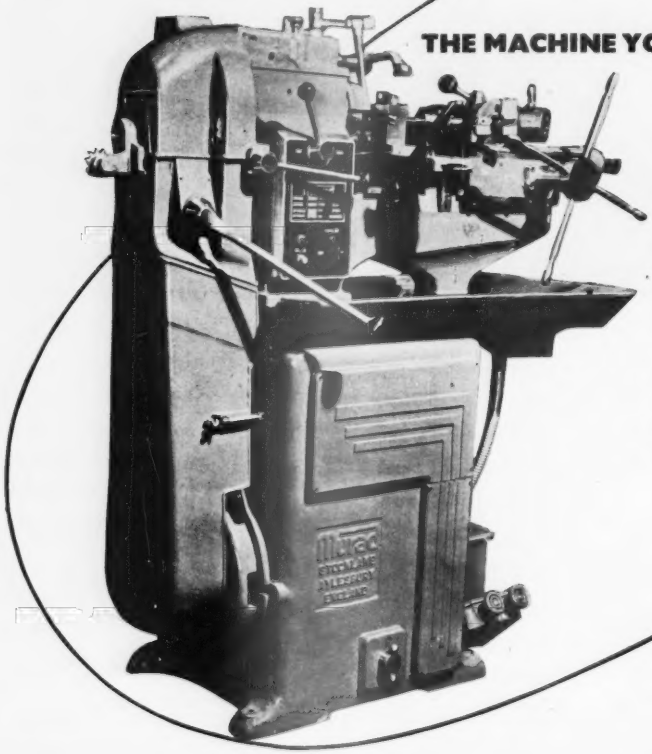
T 378

THE MACHINE YOU CAN'T BUY



We were in a hurry. Month by month
we stepped up production of our capstan lathes
and as each bottleneck appeared
we opened it.
When the internal grinding bottleneck
appeared we were "stumped."
Delivery of another grinder such as we were
using was 37 months.
So What?
Well, we set to work and designed and built
our own and it's doing its job in fine style.
Sorry! We don't propose to build
any more—not yet anyway—
too busy making capstans to meet your insatiable demand.
Later? Perhaps.

THE MACHINE YOU CAN BUY



Even if you can't buy
our grinder you can still buy
the MURAD 1" High Speed
Capstan Lathe. This is the machine
that holds unchallenged
leadership in the one-inch class
of capstan lathes.
That is why production
mounts week by week, month by
month and still we
can't catch up.
Full details
are yours on request.

Murad

MURAD DEVELOPMENTS LIMITED, STOCKLAKE, AYLESBURY, BUCKS. Phone: Aylesbury 790 (4 lines) Cables: "Muradite" Aylesbury.

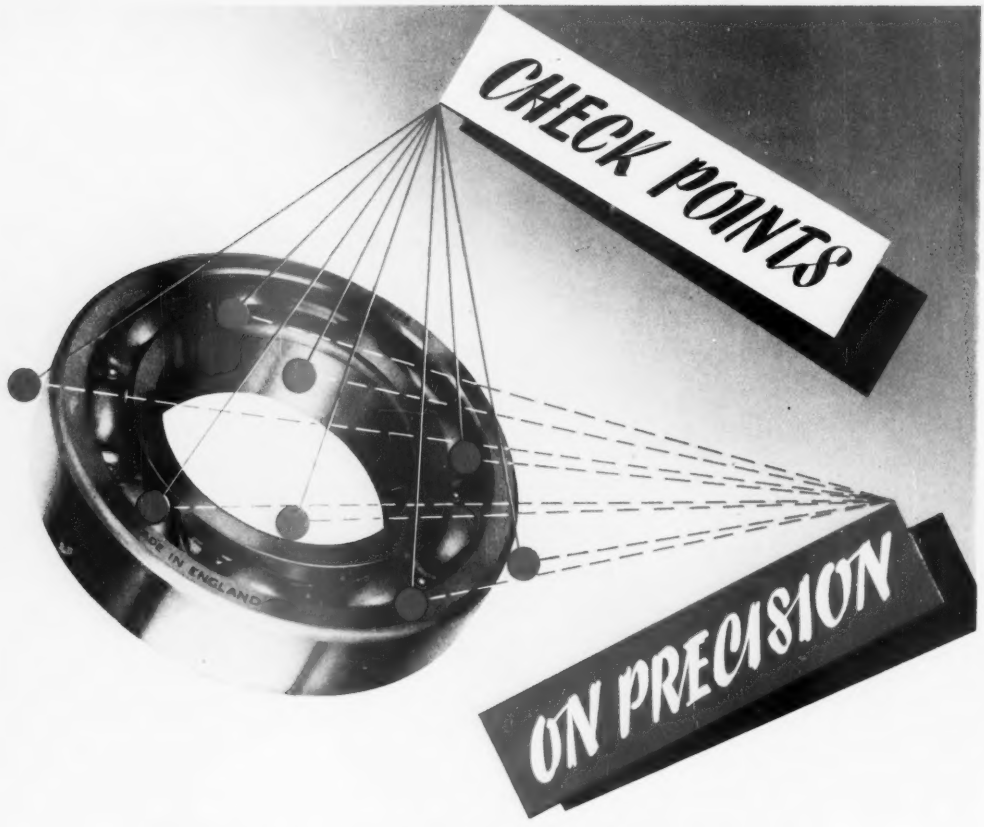
P 1802

Before selecting a material for use at high temperatures careful study of all available information must be made. That is where we can help. We have research data and service results covering the use of a wide range of materials in various operating conditions, and may already have the solution to your own specific problem. The chances are that one of our high-nickel alloys will best meet your high-temperature requirements—but ask us *before* you go ahead.

WIGGIN

FOR COLD FACTS ON HOT METALS

HENRY WIGGIN AND COMPANY LIMITED · WIGGIN STREET · BIRMINGHAM · 16



In the manufacture of R&M Bearings precision has many points of view. Dimensions must be exact to the finished tolerances not only on fitting diameters but in the clearance allowed between the balls and tracks, cages and balls, and also cages and tracks.

After measurement of individual diameters, etc., special multi-point gauges where minute errors are magnified on sensitive dials re-check one dimension against another.

Only the most scrupulous attention to machining will produce the accuracy which will pass these tests, and ensure long life and silent running of R&M Bearings.

BALL & ROLLER R&M BEARINGS

RANSOME & MARLES
BEARING COMPANY LTD.
NEWARK-ON-TRENT, ENGLAND

K.RM.96

'Cassel' Mechanised Salt Bath Furnaces

for Carburising, Heat-treatment, Austempering, Martempering and Brazing.

All the advantages of the salt bath process plus

- **Flow production at high rates**
- **Less handling and lower labour costs**
- **Uniform results**
- **Automatic control of time and temperature**
- **Elimination of fume and dust**

I.C.I. design and manufacture mechanised salt bath furnaces to meet individual requirements. For further information, or for details of 'Cassel' salts and standard types of furnace, consult—

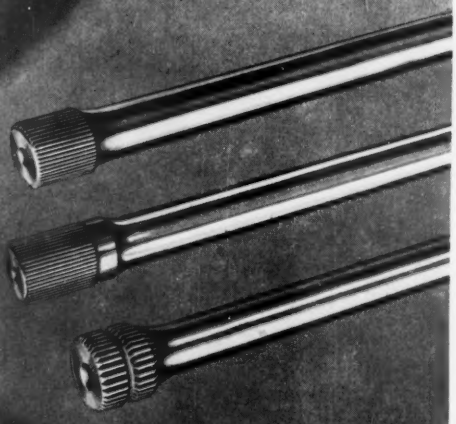
ICI **IMPERIAL CHEMICAL INDUSTRIES LIMITED, LONDON, S.W.1**

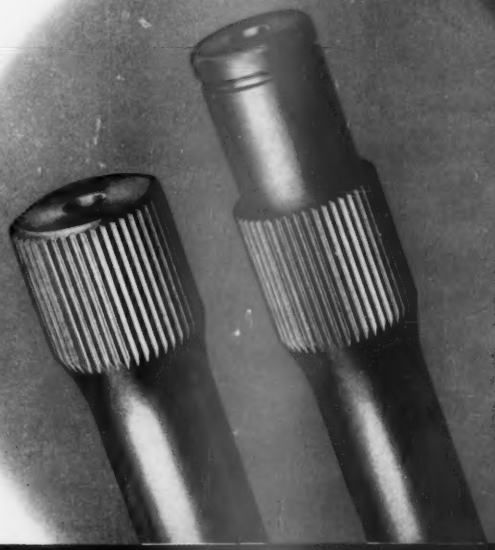
C.C.154

TOLEDO WOODHEAD SPRINGS LTD.

TORSION BARS

Toledo Woodhead Splined Torsion Bars are being supplied to several famous vehicle manufacturers, and our new plant has capacity to supply more. May we have your enquiries?





WE ARE ALSO THE MANUFACTURERS OF

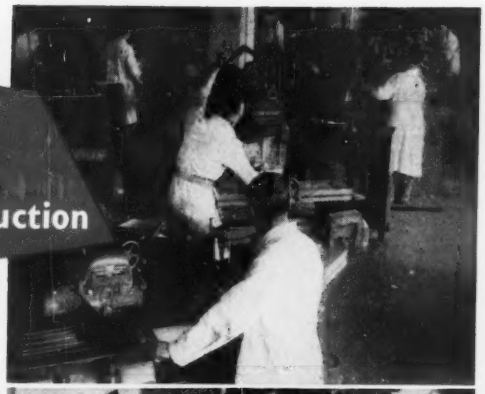
KANTLINK

SPRING WASHERS

TOLEDO WOODHEAD SPRINGS LIMITED SHEFFIELD 3

TWS 47

Increased tool life means increased production



All our Tool Steels are the result of exhaustive research and those here described are recommended as ideal for use in the modern tool room.

KE 970

This type of 2% Carbon — 14% Chrome Tool Steel is recommended for exceedingly high duty work such as the blanking of Hard Steels, Stalloy and kindred laminations in Silicon Iron and for many other tools where high duty and long service are desired. It constitutes probably the greatest advance in tool steel production since the invention of High Speed Steel. After hardening, the temper may be drawn over a much wider range than is permissible with ordinary alloy tool steels.

KE 595

A Super Oil-hardening Tooling Steel eminently suitable for Reamers, Taps, Gauges, End Mills, Broaches, etc. Supplied in the well-annealed and easily machinable condition. Possesses admirable hardenability in thicker sections, combined with minimum distortion and maximum toughness.

KE 672

Known universally as "The Blue Line Die Steel," this Oil-hardening General Tool Room Steel combines excellent hardening properties with good durability, homogeneity, ease of machining and minimum distortion.

KE 637

The perfect Low Temperature, Oil-hardening, Non-shrinking Tool Steel for Ring Gauges, etc., where final grinding is not possible and where minimum distortion is required.

Photographs by courtesy of Sharpe and Wright Ltd., Hove

Supplied in the form of hot rolled bars, precision ground bars or as forgings

LET US HELP YOU WITH YOUR TOOL ROOM PROBLEMS

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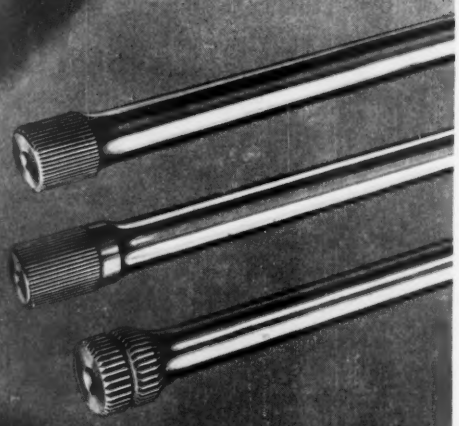
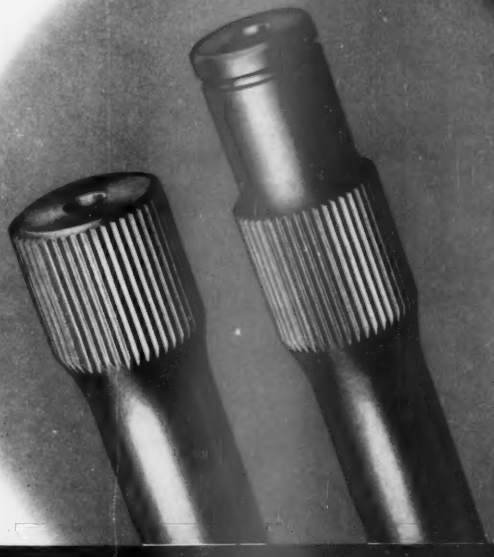
***Other specialities** include high grade High Speed and Stainless Steels as Forgings, Wire and Bars, black rolled or precision ground to the finest limits*

TOLEDO WOODHEAD SPRINGS LTD.

TORSION BARS

Toledo Woodhead Splined Torsion Bars are being supplied to several famous vehicle manufacturers, and our new plant has capacity to supply more. May we have your enquiries?





WE ARE ALSO THE MANUFACTURERS OF

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TOLEDO WOODHEAD SPRINGS LIMITED SHEFFIELD 3

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(K)(E) 595

A Super Oil-hardening Tooling Steel eminently suitable for Reamers, Taps, Gauges, End Mills, Broaches, etc. Supplied in the well-annealed and easily machinable condition. Possesses admirable hardenability in thicker sections, combined with minimum distortion and maximum toughness.

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Photographs by courtesy of Sharpe and Wright Ltd., Hove

Supplied in the form of hot rolled bars, precision ground bars or as forgings

LET US HELP YOU WITH YOUR TOOL ROOM PROBLEMS

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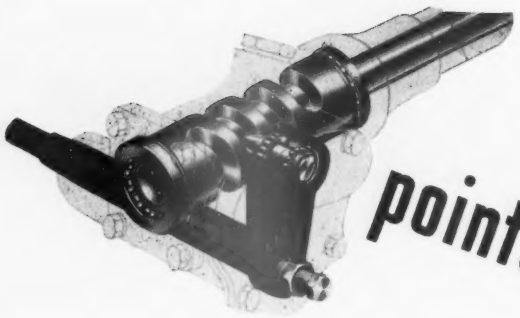
Other specialities include high grade High Speed and Stainless Steels as Forgings, Wire and Bars, black rolled or precision ground to the finest limits



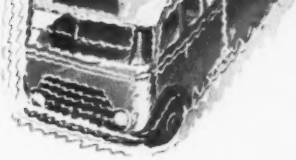
points of stress . . .

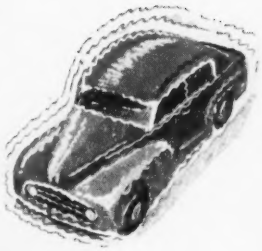
are strong points . . .

with Smethwick Drop Forgings

SDF SMETHWICK DROP FORGINGS LTD · SMETHWICK & KIDDERMINSTER ENGLAND

Andre stops vibration



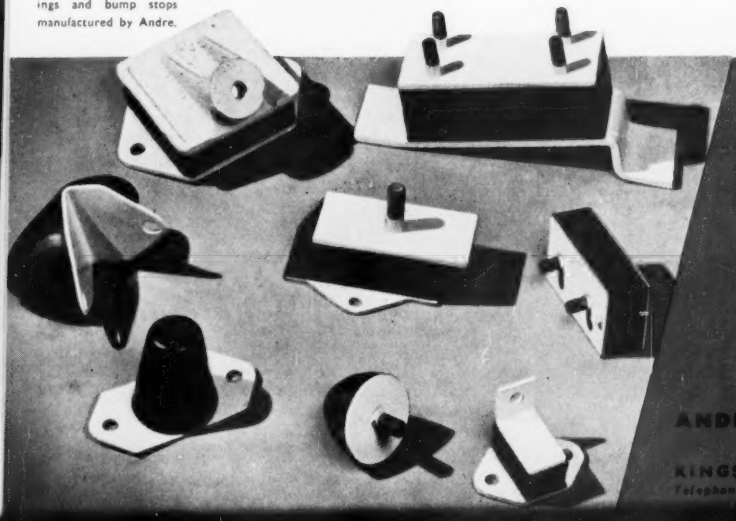


Andre RUBBER

Rubber-bonded-to-metal mountings provide the most effective and economic method of damping vibration. But the design must be right and the bond beyond question. You can be sure of both at Andre where the technical service and production facilities of a company specialising in bonding and moulding rubber are at your service. Our Technical Department welcomes the opportunity to co-operate in the development of new applications of engineering rubber.

Write for a copy of "Elastomeric Engineering Examples"—a photographic record of the wide range of rubber and rubber/metal components produced by ANDRE.

A selection of Rubber-bonded-to-metal mountings and bump stops manufactured by Andre.



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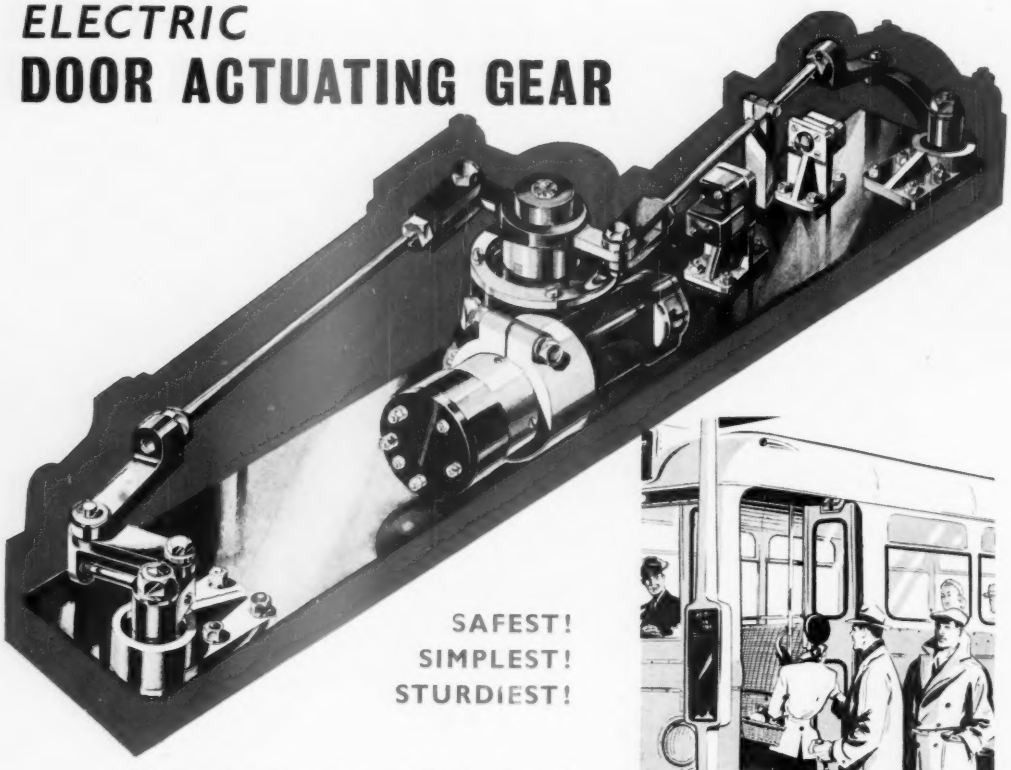
(A SILENTBLOC COMPANY)
KINGSTON BY-PASS, SURBITON, SURREY
Telephone: Elmbridge 6500/3 Telegrams: Andre, Surbiton

C.A.V.

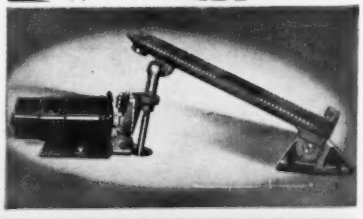
ELECTRIC
DOOR ACTUATING GEAR



SAFEST!
SIMPLEST!
STURDIEST!

In addition to improving safety, excluding draughts, and the generally recognised merits of power actuated doors, C.A.V. equipment offers the following advantages:

- Simple, robust and light in weight
- Applicable to folding or glider doors
- Compact—easily housed in normal header box
- Low initial cost
- Easy to install—no pipework necessary
- Independent of braking system
- Safety interlocking devices easily incorporated
- Low current consumption
- Slipping clutch prevents injury or damage
- Low maintenance costs



Typical safety interlock on accelerator pedal.

C.A.V.

Fuel Injection and Electrical Equipment

C. A. V. LIMITED, ACTON, LONDON, W. 3

174-499

Trade Mark granted to Daniel Doncaster 1778

. . . on calling workmen by their Christian names

It was a source of pride to the good Victorian master that he should know his workmen well and by their names, Jack, Sam or Jim. Nor was the gratification one-sided, for Jack took equal pride in knowing ' Mr. John,' not as a remote stranger, but as one to whom he could always put his views and sometimes take his troubles.

Need we in these days deplore such harmless snobbery? Certain it is that, of the five generations of Doncasters who have carried on the firm since 1778, three took pride in calling their every man by name, and every man in turn knew and respected the Doncaster of the day.

From the turn of the century, and with each succeeding decade, development in the Doncaster business of forging and treating steel became so rapid that new faces appeared more quickly than human recognition could take them in. But the spirit remains — that of a concern of individuality, character and family Quaker tradition, with a strong personal contact between management and men, augmented now by the fact apparent that there is 'room at the top.' Of the eight present directors, two are from the Doncaster family, two came from outside, and four started in offices or works as boys or young men without family connection or influence beyond their own worth.

We welcome correspondence from Careers Masters of every kind of school, technical and other colleges, and from University Appointments Bureaux, as well as from parents, on the interesting and progressive opportunities we have to offer to young men.

DONCASTERS 1778 D D

DANIEL DONCASTER & SONS LIMITED · SHEFFIELD

FORGINGS · DROP FORGINGS · HARDENED STEEL ROLLS · HEAT TREATMENT

D D

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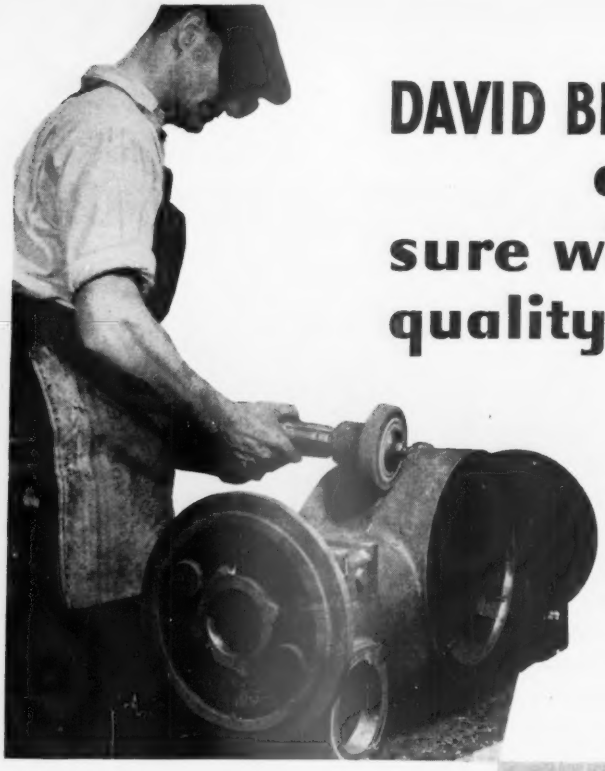
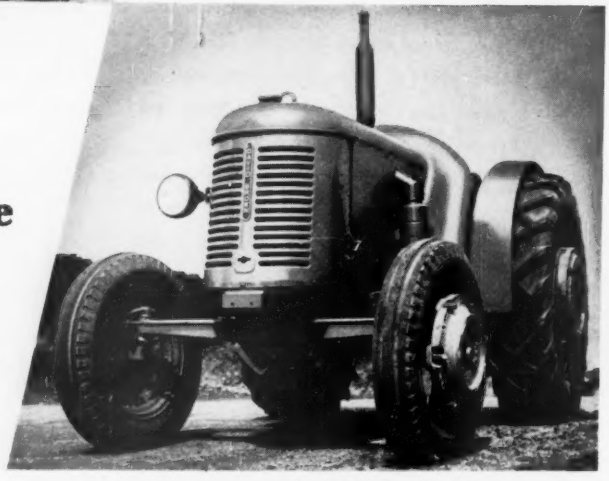


DAVID BROWN are making sure with HARPER quality

Back axles for the David Brown Tractor have to stand up to the arduous work demanded of them by the farmer. Another example of how famous firms rely on Harper quality.



Harper and **Harper-Meehanite** *Castings*

Harper quality covers grey iron and Meehanite castings, and also metal pressings, machining, enamelling and other finishes and sub-assembly work.

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ALBION WORKS Phone: WILLENHALL 124 (5 lines) Grams: HARPERS WILLENHALL **WILLENHALL**

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CASTINGS WITH THE HARPER SKIN

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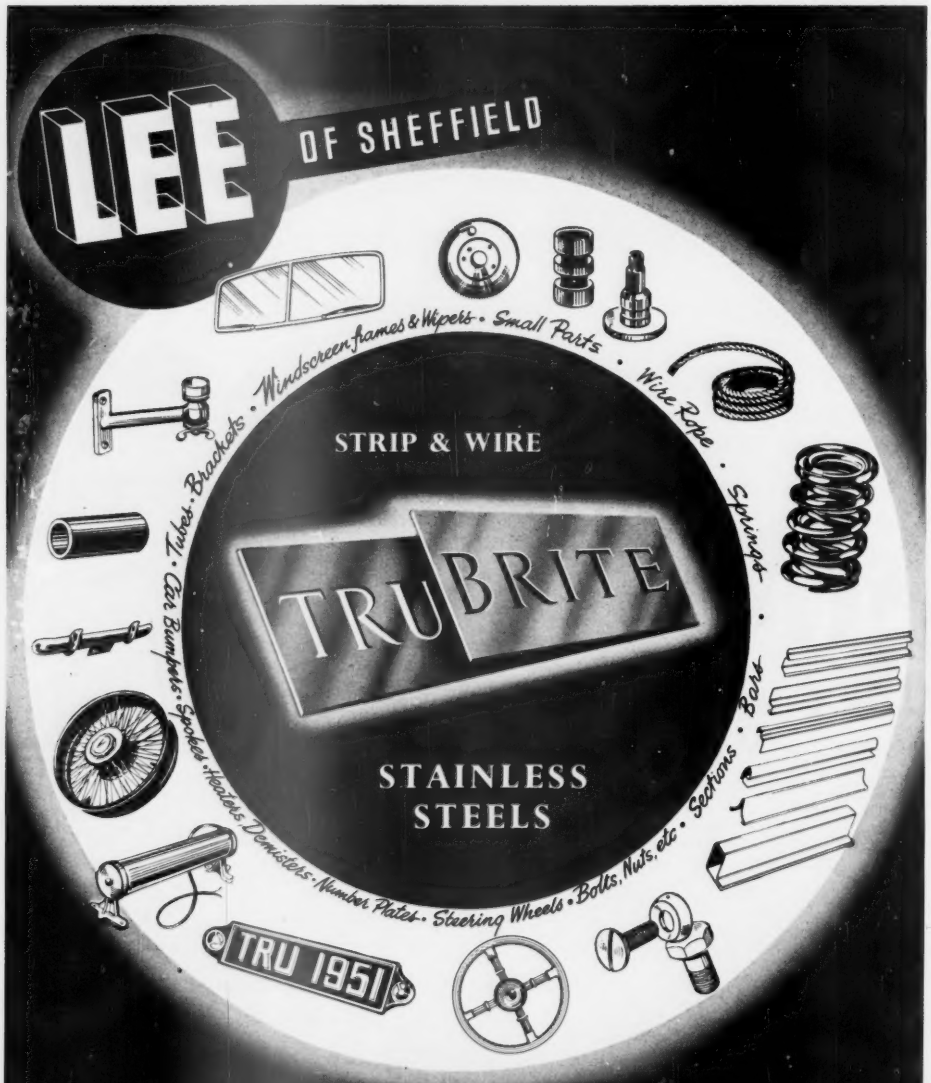
HOFFMANN BEARINGS

Increasing speeds are imposing conditions of increased severity on ball and roller bearings. But the changing conditions are being fully met by improvements to materials, design and workmanship. It is significant that Hoffmann are being used for most modern racing cars and numerous standard production models.

HOFFMANN

THE HOFFMANN MANUFACTURING CO. LTD., CHELMSFORD, ESSEX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OF SHEFFIELD
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TRUBRITE
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Windscreen frames & Wipers · Small Parts · Wire Rope · Springs · Bars · Sections · Bolts, Nuts, etc. · Steering Wheels · Number Plates · Heaters · Demisters · Spokes · Car Bumpers · Tubes · Brackets
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The Motor Car Industry is another example of the ever increasing uses of "Trubrite" Stainless Steel in the form of Cold Rolled Strip and Wire, quite apart from the already well established uses of Bright Mild Carbon Steel Strip, Bright Steel Bars and High Strain Wires, with which we have for so long retained the confidence of Motor Car and Motor Accessory makers.

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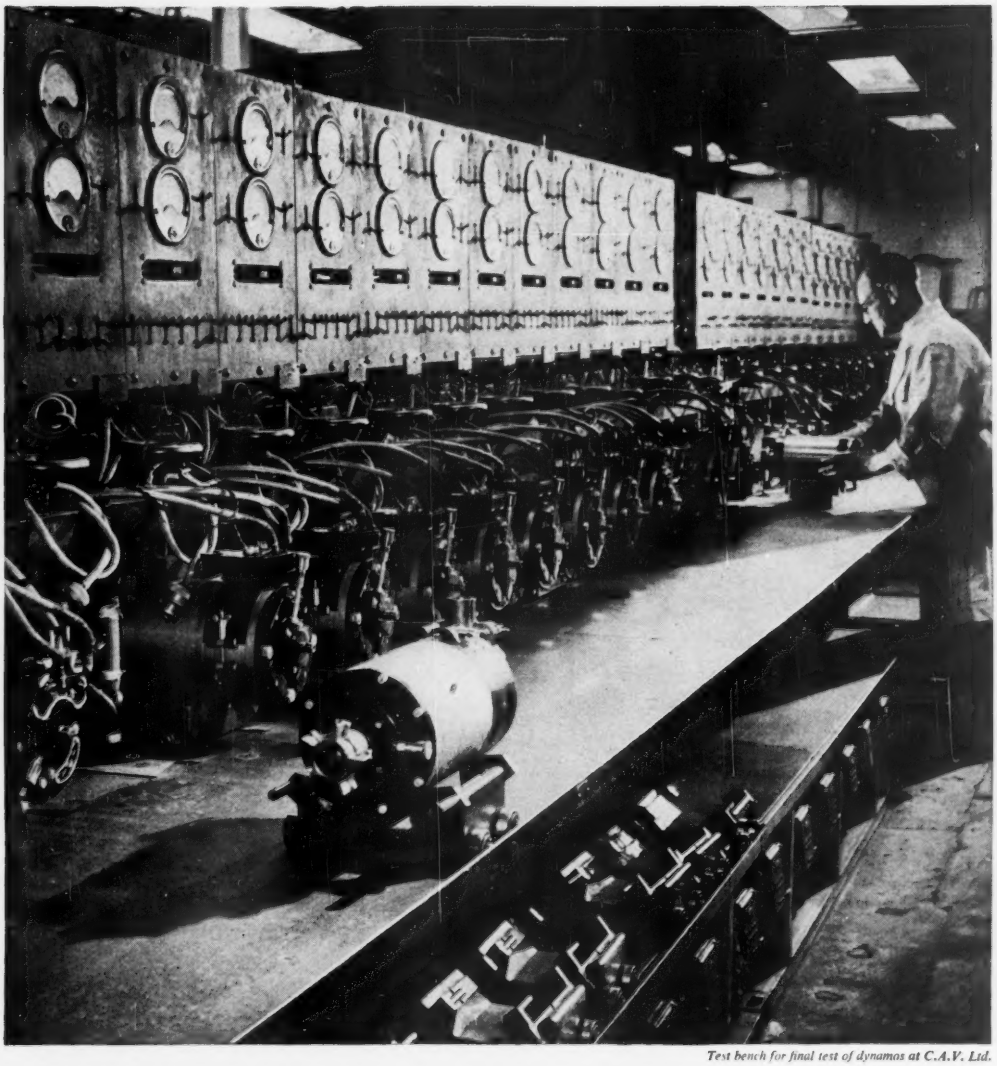

Test bench for final test of dynamos at C.A.V. Ltd.

How much time does testing take?

Testing will always be an important, though unproductive, aspect of factory production. And, because it is unproductive, it should occupy the least possible time consistent with accuracy. In your works, are your test methods as quick and foolproof as they might be? There are many improved electrical test methods which can offer you faster, more certain non-destructive testing, which can be operated by the unskilled, and which are suitable for single or multiple tests. It might pay you to investigate these ways to *make the efficient use of electricity increase productivity.*

WHERE TO GET MORE INFORMATION

Your Electricity Board will be glad to help you get the utmost value from the available power supply. They can advise you on ways to increase production by using Electricity to greater advantage — on methods which may save time and money, materials and coal, and help to reduce load shedding. Ask your Electricity Board for advice: it is at your disposal at any time.

Electricity for PRODUCTIVITY

Issued by the British Electrical Development Association

25 years on the long path to perfection

Not quickly does the master thatcher learn his ancient art. Many patient years — some would even say generations — of apprenticeship must be served before the laying of one perfect roof of straw or reed. In contrast, the art of making tubes is comparatively new. But Tru-Wel Tubes already have behind them a full generation of steady progress towards perfection, and many of their users are kind enough to call them almost perfect now.

Tru-Wel's uniformity of strength, concentricity, wall thickness and composition is *precisely* what every assembly line requires.

tested... tried... trusted

TRU-WEL

ELECTRICALLY WELDED STEEL TUBES

Tru-Wel Electrically Resistance Welded Steel Tubes

Supplied in straight lengths from ¼" o/d to 4½" o/d in gauges 22 to 6, or manipulated to your special requirements.

Tru-Wel Stainless Steel Tubes

in straight lengths ½" to 2" o/d in gauges 20 to 12 or manipulated to measure, are made from most Austenitic Nickel Chrome Steels and some Ferritic Nickel-free steels.

TUBE PRODUCTS LTD., OLDBURY, BIRMINGHAM A TI COMPANY

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224 HONING MACHINE

THE SOLUTION TO YOUR FINISHING PROBLEM!

Putting uniform surface control and dimensional accuracy within ten-thousandths of an inch on a production basis is one way to cut manufacturing costs. The Barnesdril 224 Honing Machine provides this high speed production method for controlling accurate stock removal and uniform surface finish in the same operation. Applied to either internal or external diameters, metal parts requiring accurate, smooth, surfaces and tolerances for straightness and roundness, can be mass-produced with assured quality control and a minimum of time and expense. High production rate in Honing Cylinders can be obtained.

Machine has 25" stroke and can hone up to 5" diameter. Write for our latest booklet—"The Honing Process".

CONTRACT HONING

We have the facilities and would be pleased to receive enquiries for Honing work from 1½ in. to 2½ in. dia. by 20 in. long and 2½ in. to 6 in. dia. by 44 in. long.

• *Made in Great Britain*

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Over to you

RADICON WORM REDUCERS

We can supply from stock a comprehensive range of assembled units of most of the standard ratios of 2¼, 3 and 4 RHU

THESE ARE IMMEDIATELY AVAILABLE

Quick deliveries for standard ratios up to 14 RHU can also be made and a Radicon Worm Reducer can be over to you in a matter of a few days

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A DAVID BROWN COMPANY

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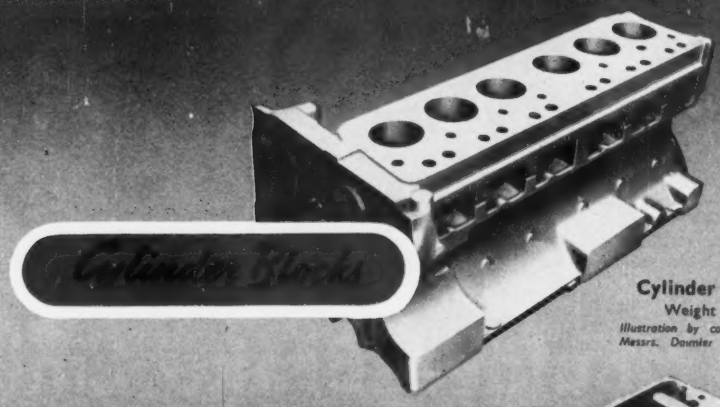
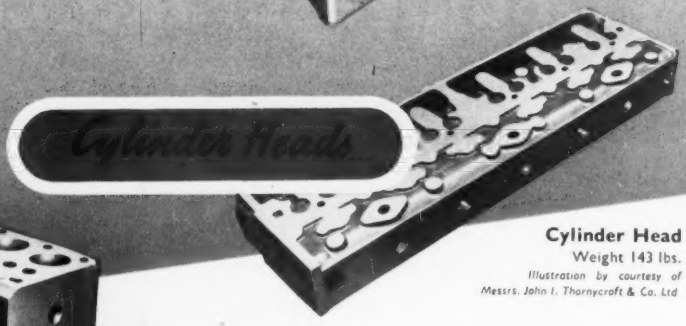
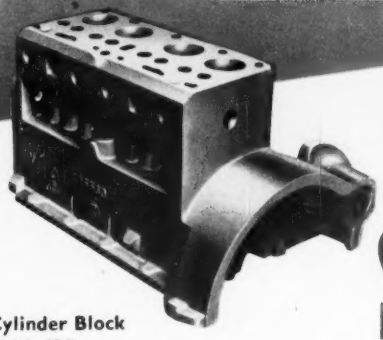
Cast Iron



Cylinder Block
Weight 633 lbs.
Illustration by courtesy of Messrs. Daimler Co. Ltd.



Cylinder Head
Weight 143 lbs.
Illustration by courtesy of Messrs. John I. Thornycroft & Co. Ltd



Cylinder Block
Weight 90 lbs.
Illustration by courtesy of Messrs. Morris Motors Ltd.

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Coventry

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Moulded Brake Linings

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- ★ Rapid 'Wet' Recovery
- ★ Extreme Resistance to Frictional Fade
- ★ Dimensional Stability
- ★ Uniform Wear to 'Wafer' Thickness
- ★ Impervious to Oil and Grease
- ★ Non-Abrasive to Brake Drums
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NON-FADE *Moulded* BRAKE LININGS

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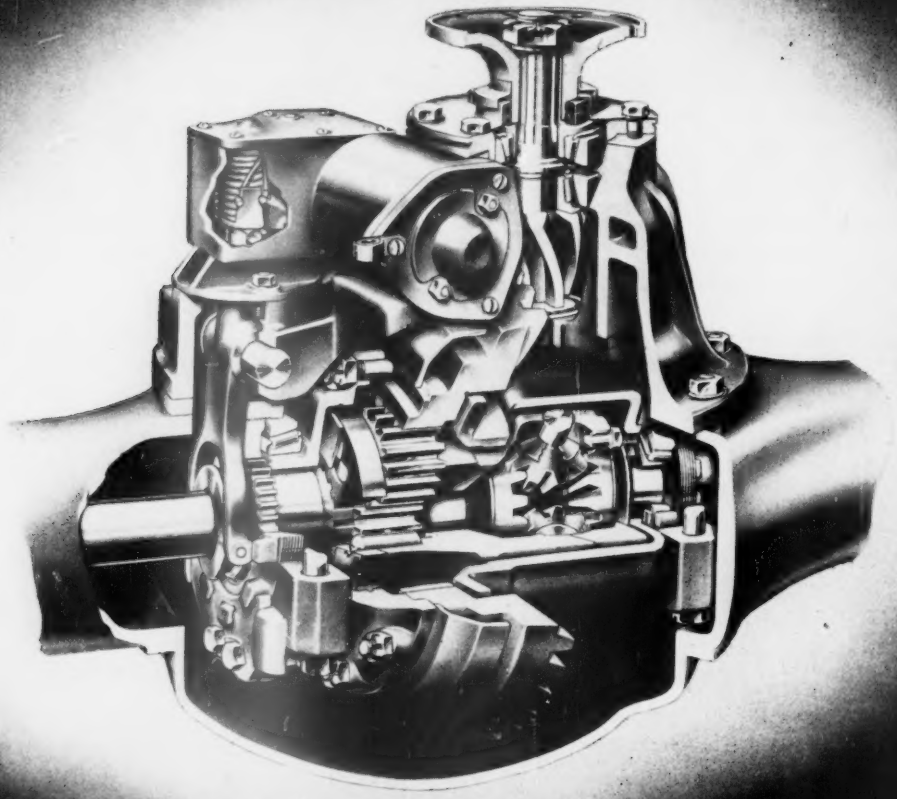
OVER ONE MILLION EATON TWO-SPEED AXLES IN TRUCKS TO-DAY

EATON

The inside story . . .

2-Speed AXLES



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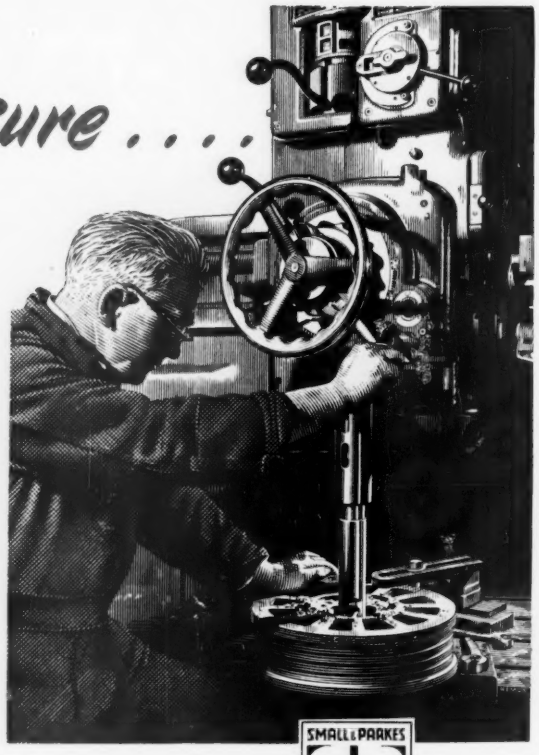

The numerous critical stages in making good clutch discs are our exclusive worry. If there were ten times as many the fact would remain that you, the user, are interested only in the final result. But because you may sometimes wonder why DON-FLEX Clutch Discs are just that little bit better, we are taking you behind the scenes . . .

Here, we are making sure of precision in production by making and maintaining our equipment in our own engineering shops.

An obvious thing to do? Precisely . . . the pains which *we* take over even the most obvious things mean that *you* can always be sure of DON-FLEX.

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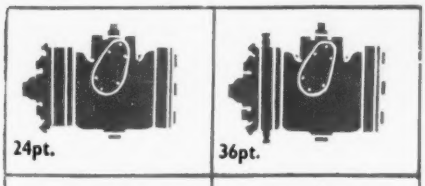
A new chassis Lubricator by CLAYTON DEWANDRE for any number of points.

Installations in London and the Home Counties, by UNIVERSITY MOTORS LTD. Lots Road, London. S.W.10



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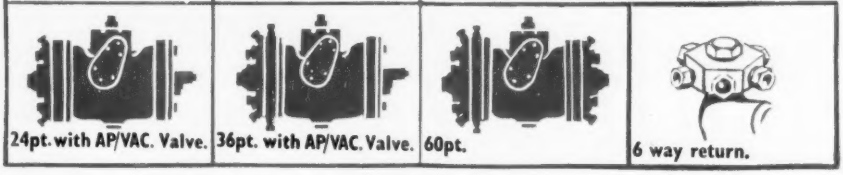

CLAYTON DEWANDRE CO. LTD.
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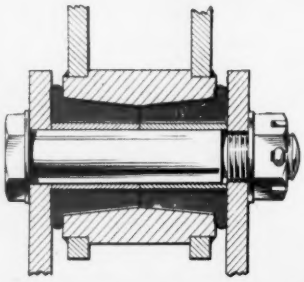
Silentbloc

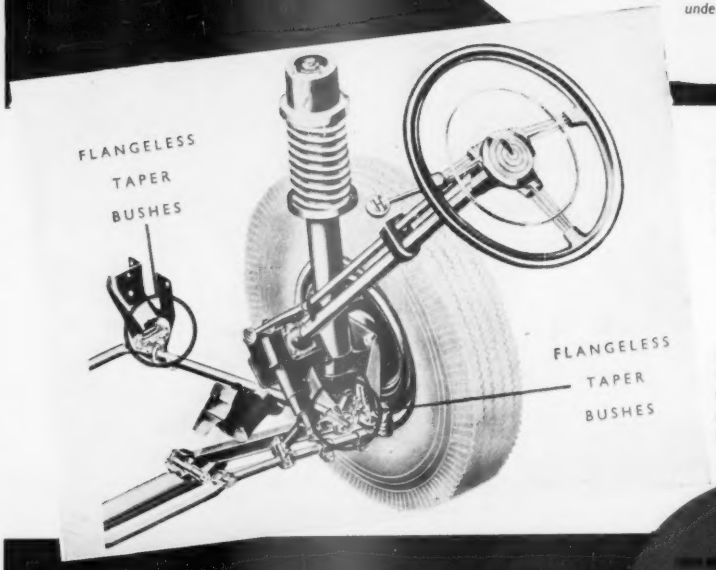
FLANGELESS TAPER BUSHES for I.F.S. systems



Scientific Development Cuts Costs

Constructed on the Silentbloc principle with a self-forming flange for extra-snug fitting and less wear these bushes ensure maximum efficiency at lowest cost.

Right: Flangeless Taper Bush sectioned to show construction. In the housing flanges develop under compression.



★ After exhaustive testing Ford Motor Company Ltd. now fit Silentbloc Flangeless Taper Bushes in the I.F.S. of the "Consul" and "Zephyr Six".

Left: Front Suspension of the "Consul" and "Zephyr Six".

Flangeless Taper Bushes and Frustacon mountings provide the first complete and scientifically designed rubber insulation between wheel and body.

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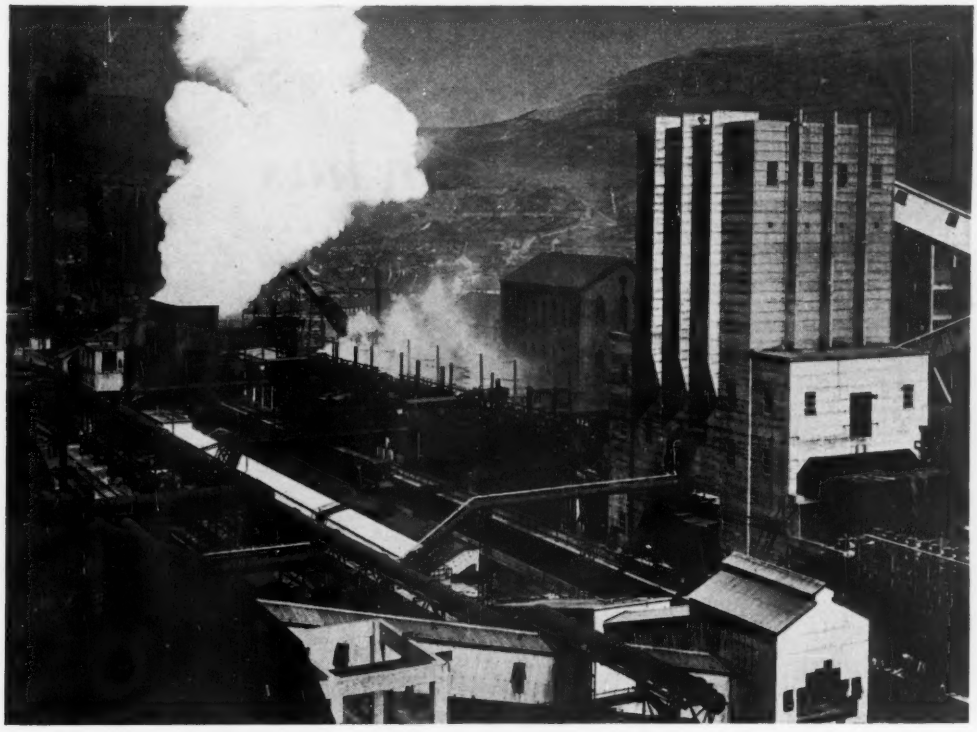
VICTORIA GARDENS
LONDON, W. 11. TEL. PARK 9821

25 YEARS



Coke Ovens at the Margam Works of The Steel Company of Wales Limited, Port Talbot

TO produce the extra coke required for the newly built and bigger Blast Furnaces at Margam Works, 90 new ovens have been installed. Together with the 54 ovens erected in 1939 these produce 15,000 tons of coke per week requiring some 22,000 tons per week of coking coal from the local Welsh Valleys. Coke Oven gas from this plant is used for heating purposes in the Margam and Abbey Works and a substantial amount is also supplied to the Wales Gas Board for use in the surrounding area.

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Ease of casting
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Accuracy of shape
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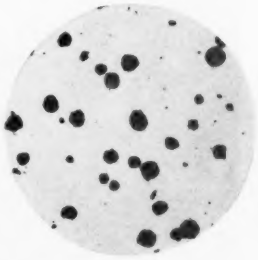
Resistance to wear
Rigidity
Resistance to corrosion

S.G. Iron has all this and Ductility too



SPHEROIDAL GRAPHITE IRON
(ETCHED) x 100

Although cast iron is such a useful material, its lack of strength and its brittleness have hitherto been serious limitations. But now the field of usefulness has been greatly extended by the introduction of a new type of cast iron in which the flake graphite, the weakening constituent, is replaced by spheroidal graphite. In the tensile test, the Spheroidal Graphite Irons have a definite yield point preceded in the stress-strain diagram by the same kind of straight line relationship as is found in steels.

Minimum properties which may be expected from three grades of S.G. iron in commercial production are as follows :—

	Maximum Stress t.s.i.	Yield Point t.s.i.	Elongation per cent.
Pearlitic	37 min.	27 min.	1 min.
Pearlitic/Ferritic	32 min.	24 min.	5 min.
Ferritic	27 min.	20 min.	10 min.

FLAKE GRAPHITE IRON
(UNETCHED) x 100

The process is the subject of patents and patent applications and the Mond Nickel Company Limited has granted a number of manufacturing licences. For the names of suppliers of S.G. iron castings write to :—

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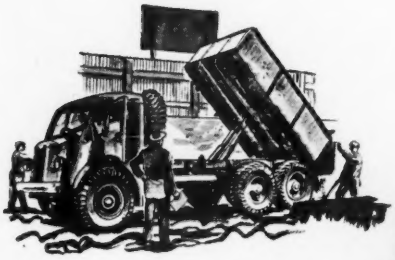

Dunlop Flexible Pipes put an end to pipe difficulties. Consider some of the practical advantages which they offer: durability and high fatigue-resistance, resulting from special Dunlop design and the traditional quality of materials and workmanship; short end-fittings that offer ease of installation; secure end-fittings that cannot be shifted by internal pressure. There are many types of coupling for original fitment or replacement, and, of course, high, medium, and low pressure ranges. The Dunlop Technical and Consultative Service is ready to advise you on all application problems.

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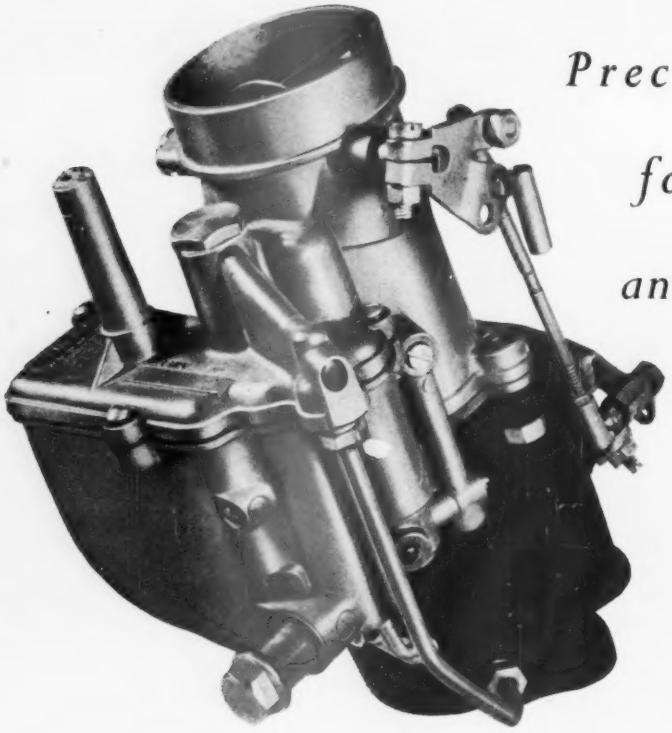
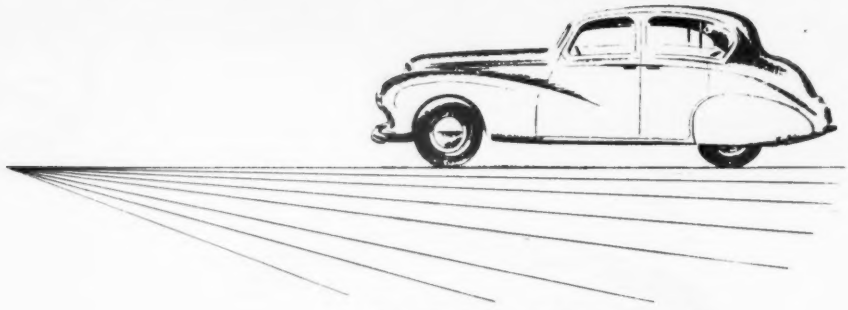


Precision Cast

for speed

and efficiency



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The Stromberg carburettor of the Sunbeam-Talbot is a fine example of pressure die casting in the modern motor car. It is of course cast in MAZAK, the metal that ensures accuracy of detail and high performance. Based on zinc of 99·99+% purity, MAZAK is ideal for the rapid and accurate production of die cast car components.

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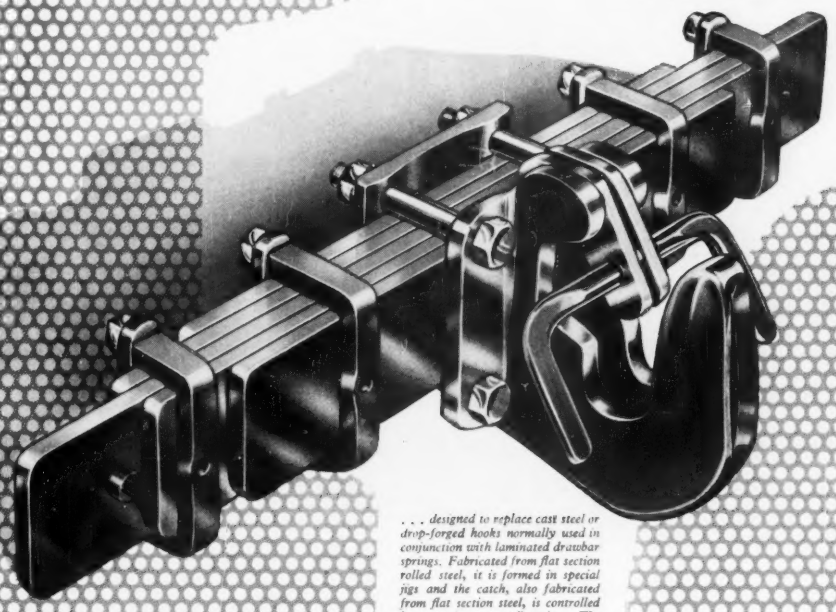
There's only one word for springs – it's

WOODHEAD



. . . designed to replace cast steel or drop-forged hooks normally used in conjunction with laminated drawbar springs. Fabricated from flat section rolled steel, it is formed in special jigs and the catch, also fabricated from flat section steel, is controlled by a tension type coil spring. The drawbar spring is of the conventional laminated 'push and pull' type and has four side grooves to locate the hook assembly to the bracing plate.

The WOODHEAD Patent Towing Hook and Spring

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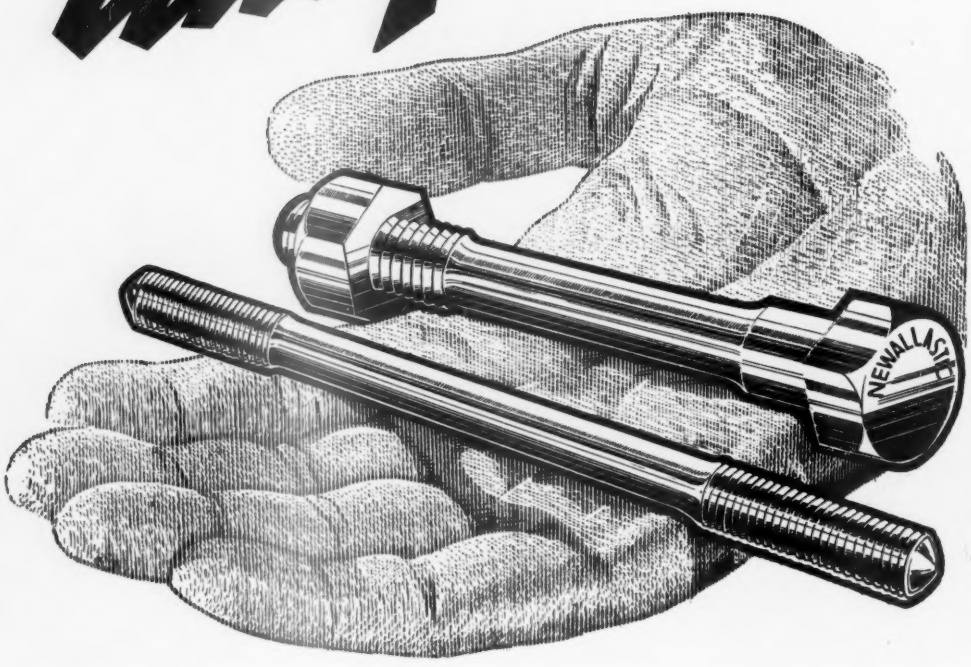
Unique



"Newallastic" bolts and studs have qualities which are absolutely unique. They have been tested by every known device, and have been proved to be stronger and more resistant to fatigue than bolts or studs made by the usual method.

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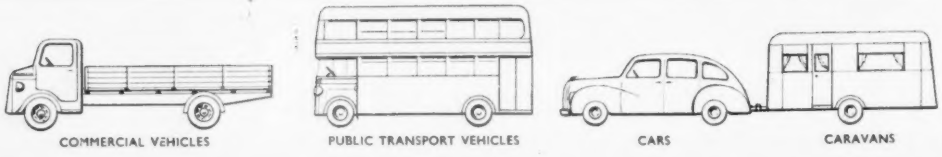

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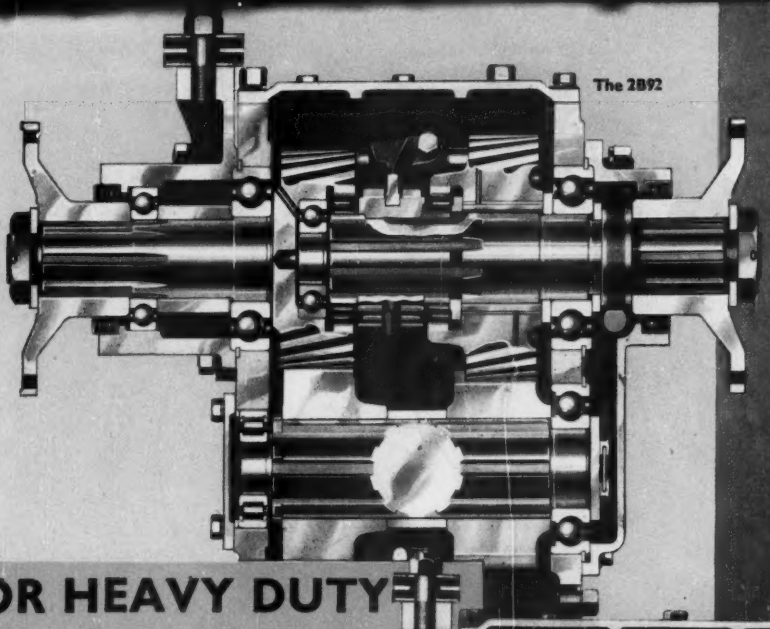
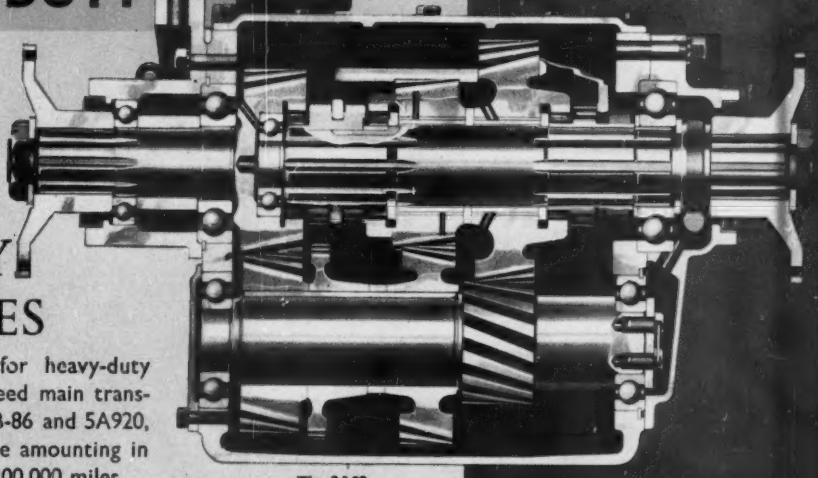


The 2B92

FOR HEAVY DUTY

TWO & THREE-SPEED AUXILIARY GEAR-BOXES



The 3A92

These gear-boxes are built for heavy-duty behind large four and five-speed main transmissions such as the Fuller 4B-86 and 5A920, and are fully proved in service amounting in many instances to 200,000 to 300,000 miles.

Conventional trunnion mounting or front stud mounting optional. The 3A92 can be equipped with the improved top-mounted power take-off, replacing the standard flat top cover, or with a power take-off gear provided on the countershaft to drive a side-mounted power take-off.

These boxes have helical gears mounted on long bushings, all changes being made by jaw-type clutches.

Gear ratios :

3A92 —	Overdrive	.754:1
	Direct	1:1
	Reduction	2.09:1
	Weight	350 lb.
A92 —	Direct	1:1
	Reduction	2.30:1
2B92 —	Direct	1:1
	Reduction	1.313:1
	Weight	315 lb.

Exclusive European Representatives

AUTOMOTIVE PRODUCTS COMPANY LTD.,
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Fuller

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use G.E.C.

ELECTRIC Process Heating

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● Bold thinking about bigger production demands new methods of heat treatment. G.E.C. *electric* process heating ensures maximum output and true economy. The many heating equipments in its range are clean, safe, simple to operate and heat rapidly to constant correct and controlled temperatures. Its versatility is being proved every day.

● Its applications are many: for heat treating steel, setting resin glues, drying paint, and numerous other processes. One example, is G.E.C. carbo-nitriding equipment, of use in the Aircraft and Motor industry for case-hardening steel.

● G.E.C. has had many years' experience in the design and construction of heating plant. The Company's specialists welcome opportunities to discuss potential uses with engineers and metallurgists.

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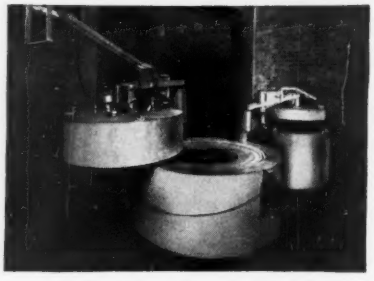

G.E.C. CARBO-NITRIDING EQUIPMENT

This equipment gives freedom from distortion and uniformity of hardened case in the charge, with consequent improvement in wear resistance.

Carbon and nitrogen are introduced into the surface of the steel at a temperature normally between 800°C and 875°C. The steel is then cooled at the rate required to give the desired properties. This plant can be adapted to many types of charge by varying the carrying arrangement inside the enclosures.

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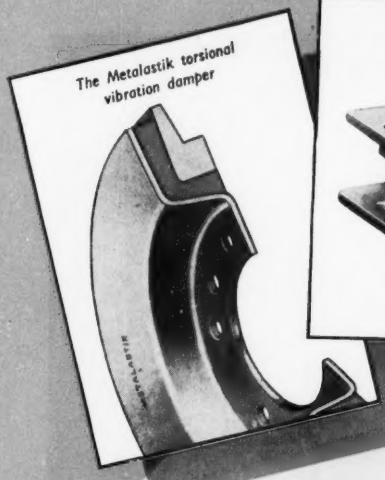

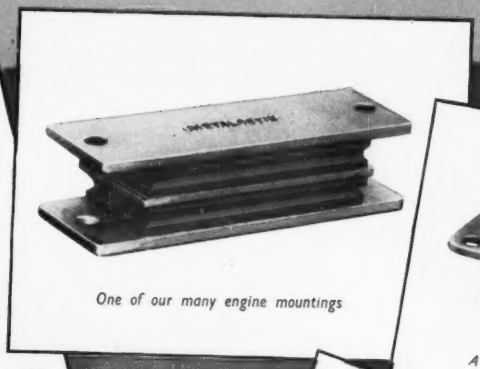
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The Heavy Duty shackle pin

Making things SMOOTHER

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Metalastik torsional vibration dampers — outstandingly simple and free from mechanical friction, have a wonderful record for effectiveness in reducing vibration and eliminating crankshaft breakage.

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Metalastik Heavy-Duty shackle-pins eliminate metal-to-metal contact and avoid the need for lubrication. More than a quarter of a million are in use, with outstanding success.

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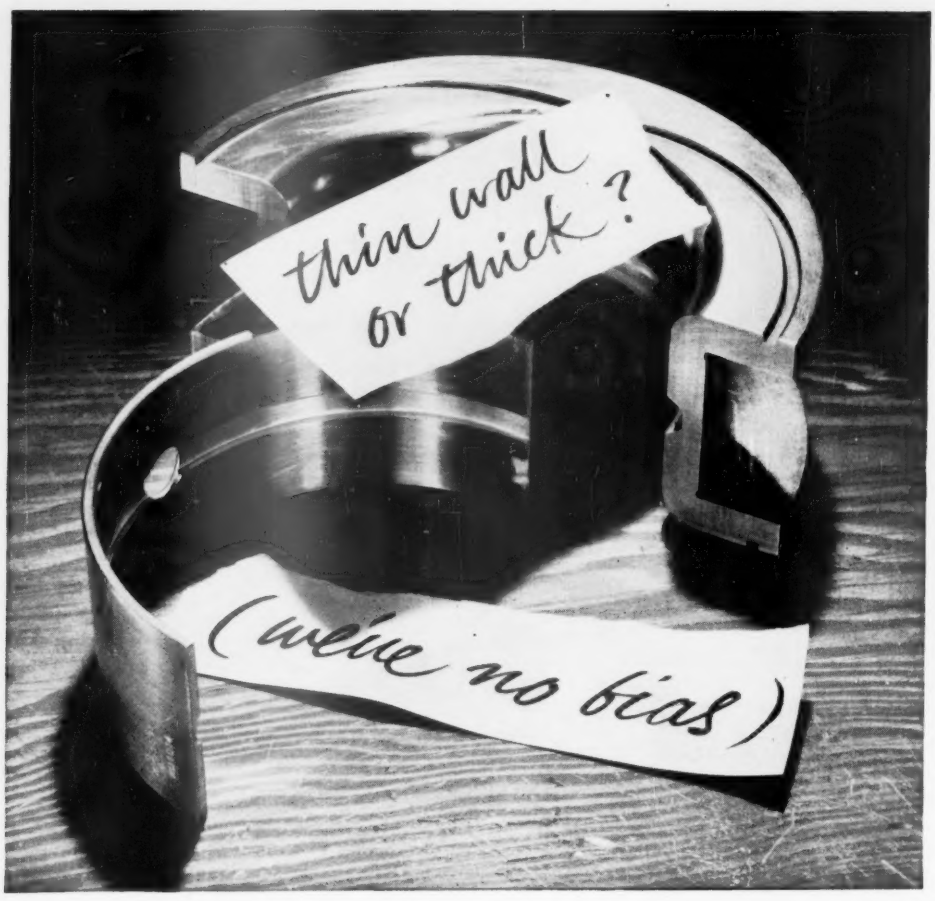

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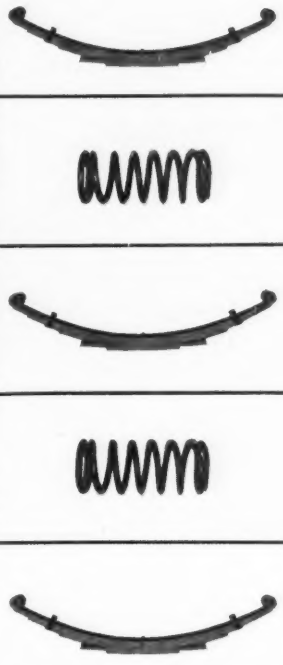

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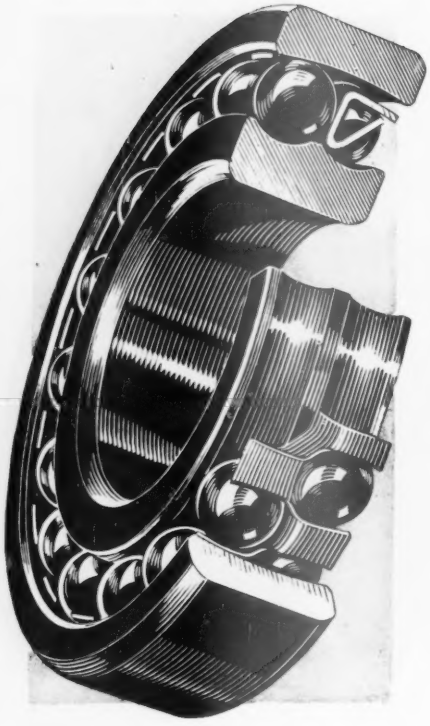

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AUTOMOBILE ENGINEER

Design, Materials, Production Methods, and Works Equipment

Editor: J. B. DUNCAN

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Engine Deposits

ALTHOUGH it is generally recognized that engine deposits have an adverse effect on engine efficiency, there is probably insufficient realization of (a) the degree of power loss that deposits can cause, and (b) the manner in which engine design influences the losses. In this connection, some investigations recently carried out in America to determine how much power loss is occasioned by deposits have considerable interest. In any given engine the deposit power loss is determined largely by the operating conditions, which it is not practical to control fully.

The deposits are, of course, products from the fuel and the lubricant. However, although the compositions of the fuel and the lubricant can affect power loss, there is now little significant difference in this respect between the majority of commercial grades. Research is now being carried out on the fuel lubricant phase of the deposit problem, but there is no evidence of any developments that will be commercially applicable in the near future. For the time being, therefore, deposit power losses can be reduced only by changes in engine design.

Investigations already carried out have clearly proved that certain combustion chamber designs are much better than others in minimizing deposit power losses. The available evidence also suggests that the designs giving the better results at present will be more favourably affected by any improvements that may be made in fuels and lubricants.

A series of tests carried out under carefully controlled conditions showed significant differences between engines of different design. One rather remarkable fact that emerged from these tests is that there is no direct correlation between power loss and deposit weight per cubic inch of engine displacement. On the contrary, an L-head engine showed a 10 per cent power loss for 0·3g of deposit per cubic inch of engine displacement, whereas an overhead valve engine showed only three per cent power loss although the deposit ratio was one-third greater. A second design of overhead valve engine with a similar deposit displacement ratio showed five per cent power loss.

Data have also been obtained for the average loss in acceleration as a result of the deposits. In these tests, two designs of L-head engines showed average losses in acceleration 2·2 and 2·5 times greater than the average losses for an overhead valve design. In fact, the investigations have shown that the effects of deposits on power losses are three times as great in some current engine designs as in others.

There seems to be no doubt that two-thirds of the power loss due to deposits is the result of a reduction in airflow caused by physical and thermal restrictions. In most automobile engine designs the danger of physical restriction is negligible. It could however, become a serious factor in designs with compact combustion chambers and large inlet valves. In such designs care must be taken to provide adequate clearance between the wall and the inlet valve to ensure that normal deposit build-up will not restrict the flow of the charge.

Thermal restriction of the charge owing to deposits is a much more potent factor. It occurs in differing degrees in all engines. The deposits cause thermal restriction by transferring heat, absorbed during the combustion and exhaust portions of the engine cycle, to the incoming charge and thereby reducing the weight of the charge that enters the cylinder. It may therefore, be reasonably assumed that the amount of surface area to which deposits can adhere may be an important factor in affecting the power loss due to deposits. In fact, tests show that there is a general tendency towards an increase of power deposit loss as the surface area engine displacement ratio increases. For example, L-head engines have a higher ratio than overhead valve engines and usually suffer greater power loss.

However, the surface area engine displacement ratio is not the only factor involved. This was proved by tests on two overhead valve engines of different design but having approximately the same ratio ; the power loss on one was 2·5 times greater than that on the other. An important factor is the location of the deposits in relation to the incoming charge, and tests show that deposits in the head cavity have the greatest effect per unit weight. Such deposits comprised only 30 per cent of the total weight, yet caused 45 per cent of the power losses. In contrast, deposits on the block, piston and valves comprised 40 per cent of the total weight but were responsible for only 25 per cent of the power loss.

These tests show that combustion chamber design is probably the most important single factor in determining engine efficiency in service. Differences between designs may not be apparent in new engines, but as service proceeds and deposits build up there may well be appreciable differences in power losses. The design features to be incorporated to minimize power loss through carbon deposits are :—

(1) The charge must be introduced into the cylinder in such a way that there is minimum contact with the deposit-coated surface ; otherwise it will be heated and there will be a reduction in volumetric efficiency.

(2) There should be adequate open areas adjacent to the inlet valve to ensure that deposit build-up will not cause physical restriction to the flow of the charge.
(3) The combustion chamber surface area/engine displacement ratio should be low.

Fuel charges now account for a much higher proportion of total operating costs than they did before the war, and all the available evidence suggests that the prices of fuel will tend to rise rather than fall. As a result, fuel costs are now a very important factor so far as sales are concerned. It is therefore essential that the possibility of increasing engine efficiency by reducing deposit power losses should be very carefully studied.

Graduates and Engineering

GRAVE doubts are expressed in some quarters as to whether the automobile industry is attracting university-trained entrants either in sufficient numbers or of the desirable quality. Indeed, similar doubts are expressed in connection with engineering industry in general. If these views are correct, it is important to investigate why engineering as a career is failing to attract men with reasonably high academic qualifications. It is also necessary to consider what type of men are required.

In this connection the views of Lord Hives, chairman and managing director of Rolls-Royce Ltd., must command respect. His opinions were expressed in an address entitled *What Industry expects from the Universities*. In the course of this address, Lord Hives said :—

"We do not expect the universities to turn out ready-made technicians, designers, metallurgists or chemists. They cannot do so. We do expect them to turn out men who are capable of thinking—men who have received the basic training in the fundamental sciences essential to their future work ; whose brains have been trained to tackle problems ; men, who, of their own initiative can follow a problem through, step by step, logically taking advantage of previous experience on allied work."

Lord Hives has certainly stated the rub of the problem succinctly so far as the type of man is concerned. A mind that has been trained to think logically and to see a problem in relation to its context and not as an isolated phenomenon should be the hall-mark of a university-trained man. Such a man, if he has a sound knowledge of engineering fundamentals, can quickly adapt himself to the special needs of whatever industry he enters.

If men of this type are to be attracted, two questions must be considered : what are the immediate and what the long term prospects? Frequently, the immediate prospects decide. There is no doubt that the man proceeding straight from the university into industry is a learner who has to be taught to apply his academic knowledge. So far as the employer is concerned, his immediate value is less, and usually much less, than that of men of the same age who have had the more usual training, comprising practical work and technical college studies carried out concurrently.

Unfortunately, many graduates consider that the possession of a degree entitles to a much higher initial rate of pay than will be warranted by the services they can perform until their academic training has been supplemented by practical training in industry. That an employer should justifiably attempt to correlate the wage or salary with the economic value of a man's services is considered normal for all men without academic qualifications, but many newly graduated men think that different standards should apply to them. This is one reason why many graduates refuse to enter industry. It is probable that industry is not the loser through this.

Long term prospects are necessarily unpredictable. To begin with, the number of senior posts carrying both great responsibilities and high remuneration, is limited. At the same time, competition for them is keen, and there is no guarantee that technical competence alone will be sufficient to fill them adequately. In fact, for many of the higher appointments in industry, character is of greater importance than purely technical qualifications.

Although it is important that the automobile industry should be able to attract men with academic qualifications, this should not be done at the expense of others. There has been in recent years a tendency, fortunately not widespread, in some organizations to exclude all except graduates from appointments to the highest rank of technical executives. There should be no bar other than lack of ability to prevent a man from reaching a high position. If an increasing number of university-trained men enter industry, it is only to be expected that eventually they will hold a more than proportionate number of the highest positions. Should this not occur it would be evidence that the universities are failing in their purpose. Nevertheless, in the past, industry has drawn much of its strength and inspiration from men who entered it early and with little academic training. It can, and should, benefit in the future from other men with the same background.

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AUSTIN '7' STRUCTURAL DESIGN

A Body-chassis Structure Unique Among Quantity Produced Cars

THE structural design of the Austin '7' is unique in that it is the only true chassisless vehicle in large quantity production. By *chassisless* is meant a vehicle structure that does not incorporate any semblance of a chassis frame. All other cars, that do not incorporate a separate frame, and which are in large quantity production in this country, have at least a box-section frame extended forwards from the body to carry the front suspension and the engine. This type of construction, where some form of frame is integrated with the body structure, is best termed *unitary*, or *integral construction*.

Much credit is due to the Austin Motor Company for their far-sightedness in venturing into hitherto unexplored fields of design in order to obtain greater structural efficiency than has been achieved before. Because of its efficiency, this type of construction is likely, eventually, to be universally adopted. This may, however, only be true of the relatively distant future, since there is much to learn about the technique involved in the use of thin gauge sheet metal in motor vehicle structures.

Generally speaking, the structure consists of two side members, extending the whole length of the vehicle, between which there are three main cross members to take the vertical load. There is one other cross member at the front which, together with the two additional front side members, forms a portal frame to take side loads at the front end. The ingenious feature of the design is that almost all these members are essential body components and are in no way additions to give strength to the vehicle. For example, the rear cross member carries the rear spring shackle and is formed by the boot floor, which is stepped down at this point. The second cross member from the rear is formed by the heel board, and carries the front eyes of the rear springs. Further forward the toe board transfers the vertical loads from the front suspension outwards to the side members. In addition to its functions as part of the structure, the cross member at the extreme front end also forms the front apron and radiator support.

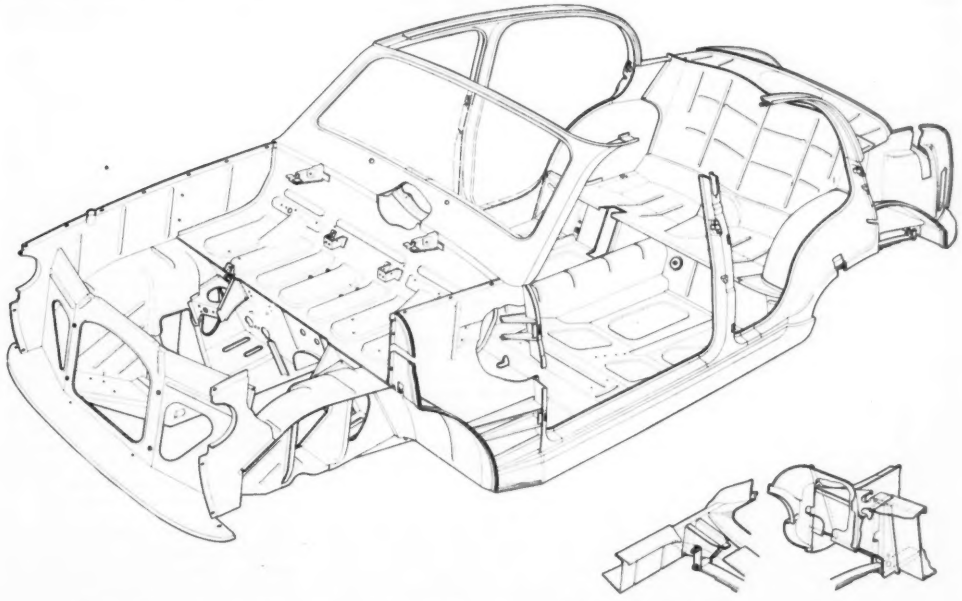
Only in the boot can the two side members be described as additional to the normal body panelling. In this compartment each member forms an inner panel, with its lower edge spot welded to the boot floor and the rear wheel-arch, and its top edge to the boot lid opening. Forward of the boot, this member forms the rear quarter trim panel, which is connected by the wheel-arch and rocker panel to the sill.

As can be seen from the illustration, the sill section is a complex box shape. At its forward end, it overlaps and is extended to the extreme front edge of the dash side. Throughout its length the sill comprises an inner plate reinforced by an outer portion that is shaped to accommodate the lower edges of the doors and to conform with the exterior styling of the body. The dash side is extended over the top of the wheel-arch and spot welded to the front bulkhead, on which the radiator is mounted. At its lower edge it is spot welded to the wheel-arch, while at the wing crown line the upper edge is similarly attached to the bonnet surround.

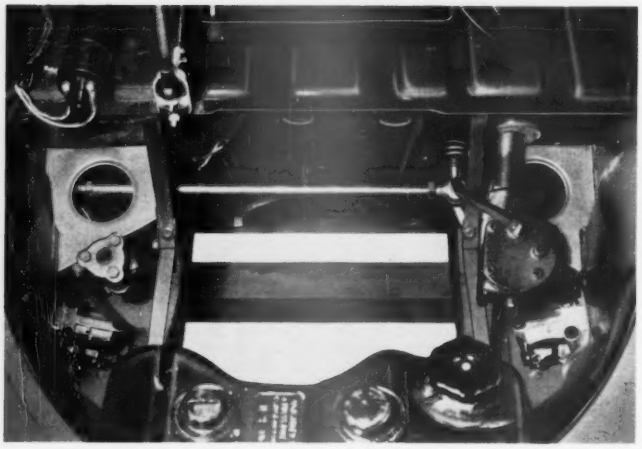
The portal frame

The portal frame is not intended to take any major vertical loads. For this reason, it is not a box section, but is a wide and shallow channel, with $\frac{5}{8}$ in flanges turned downwards at the edges. The side members are $5\frac{1}{2}$ in wide and the front component is about 6 in wide at the centre. Both the side members and the front members are of 16 s.w.g. En 2A. They are spot welded together where they overlap. The base of the frame is securely attached to the body floor.

In side elevation, the side members are not straight. From the front, they



Chassis-body structure. Except where otherwise stated in the description, all sheet steel components are of 20 s.w.g.



The steering box and idler lever may be interchanged for left or right-hand drive

slope downwards to a convenient level for carrying the wishbone bearings. Above these bearings, they are horizontal for a length of about 12 in, and then slope upwards to meet the junction between the foot ramp, on the front floor, and the toe board. Because of this angularity of the member, tensile stresses, due to lateral bending forces induced in its flanges, will tend to straighten it out, while compressive stresses, arising from these forces, will tend to kink it further. It is therefore necessary to support the flanges. This is a simple matter at the outer edges, where the member is spot welded to the vertical face of the wheel-arch, but the inner flanges must themselves have sufficient inherent strength because there is no vertical member to which they can be conveniently attached.

For this reason, an L-section plate is spot welded to the inner face of the down-turned flange of the side member. This gives the member an overall depth of 2¼ in at the front suspension attachment point. As the L-plate is carried forward, it is reduced in depth until it is just deep enough to carry the bumper irons. These irons are bolted between the L-plate and a lug welded underneath the overlap of the side member and the cross member. From the suspension attachment point, the lower edge of the L-section plate is continued straight through to the rear. The overall depth is therefore considerable under the forward part of the front floor ramp. This depth is perhaps greater than necessary if structural strength alone is considered. It has been adopted, however, for mounting the brake and clutch pedals. The stems of these components, forward of the toe board, are passed through slots in the flat upper surface of the side member.

The attachment of the side member assemblies, at the root ends of the portal frame, is very sound. The upper face of the channel is spot welded along each edge to both the ramp and, further back, to the front floor. A large lightening hole is cut out of the centre of the channel beneath the front floor, and the member ends where the floor is stepped up to form the front seat brace. The horizontal bending moments at the root of the portal frame are therefore taken by a beam, composed of the body floor and ramp, with its edges stabilized against buckling at the rear, by the front seat brace and, at the front, by the toe board. In order to counter the forces tending to tear the inner edge of the side member away vertically from the junction between the toe board and the ramp, an 18 s.w.g. angle-section tie is welded down the toe board and the L-section plate. At the outer edges of the members, this function is, of course, performed by the wheel-arch.

Front suspension mounting

Although, as has already been stated, most of the side loads from the front suspension are taken on the portal frame, a certain proportion of these loads, applied through the upper transverse link, are distributed partly to the portal frame and partly to the top of the wheel-arch. This distribution of load is effected through a vertical channel pressed into the side of the wheel-arch and suitably reinforced. The channel acts as a beam with its lower end attached to the portal frame and its upper end to the top of the wheel-arch. So far as these loads are concerned, the top of the wheel-arch forms another beam mounted between the front bulkhead and the junction of the dash and toe board. The path of the load from the upper transverse link is through a 14 s.w.g. pan-shaped pressing, on which the shock absorber is mounted, to the sides of the channels into which the pressing is spot welded.

Reinforcement is provided for the channel sides by 18 s.w.g. doubling plates, with flanged inner and outer edges. The doubling plates extend approximately 1 in outboard of the vertical face of the wheel-arch. This reinforcement is necessary because the channel cannot be pressed to sufficient depth to provide a beam strong enough to carry the loads. There is a hole in the inner wall of the channel to clear part of the shock absorber. The side loads from the lower transverse link, or wishbone, are taken directly to the portal frame through the lugs that carry the wishbone bearings. These lugs are formed by a channel section extending transversely across the under surface of the side member and welded to it, and are appropriately cut away to clear the wishbone arms as they move up and down.

Vertical loads are taken entirely through the inner face of the wheel-arch. They include the vertical components of the loads at both the upper and lower transverse link attachment points, as well as those from the spring and the bump and rebound stops. The inner edges of the lower bearing lugs are welded to the L-shaped reinforcing plate on the inner flange of the side member. At their other ends, they are welded adjacent to the outer flange of the side member. It is through these welds that the vertical components of the lower bearing loads are carried into the side member, the greater part being taken at the inner edge. Each pair of lugs is offset outboard of the adjacent side face of the vertical suspension channel. The vertical loads from the other parts of the suspension system are, of course, transferred directly to the side walls of the vertical channel through the pan-shaped pressing that carries the spring, shock absorber, and bump and rebound stops.

From the suspension channel, there are two paths that the loads may take. One is directly to the rear through the vertical face of the wheel-arch to the toe board and thence outwards to the main side members of the structure. The other is through the forward portion of the vertical face of the wheel-arch to the front bulkhead and out to the forward extension of the main side member over the top of the wheel-arch. There is no doubt that both paths are followed simultaneously, the whole system forming what is known as a *redundant structure*. Torsional loads on the whole vehicle, due to bumps, etc., applied to one wheel, are transferred across the structure at the front end by both the front bulkhead and the toe board, and are diffused into the box enclosed by the wheel-arches, side members, and bonnet surround. The form of bonnet opening adopted is one that detracts the least possible amount from the strength and stiffness of this box.

Steering brackets

Loads on the steering unit are very complex. They are difficult to analyze because of the offsets due to the drop arm and its spindle, and because of the large variations possible in the

angular relationships between the track rods and side rods. In spite of this complexity, these loads can be experimentally assessed with reasonable accuracy; but the principal difficulty is to find a strong enough part of the structure on which to mount the unit. In this case, the only convenient part is the vertical suspension channel but, because of its relatively small depth at the height at which the steering unit must be mounted, there is insufficient space in which to make a welded joint strong enough to hold the bracket. For this reason, a bolted-on stay has been added to tie the bracket to the lower part of the suspension channel where it is strengthened by the spring pan.

As can be seen from the illustration, the bracket is a complex shape, providing a flat surface on which the steering box is bolted. Three bolts are employed. They are screwed into a tapping plate under the bracket. This bracket is spot welded to the side of the wheel-arch and to the toe board, and is arc welded to the suspension channel. Two brackets are fitted, one to each side of the vehicle, to accommodate left or right hand steering. The bracket on the side opposite to the steering wheel carries the idler lever.

A sectioned view of the front end structure

Front end structure

The dash panel is spot welded to the top of the toe board. It extends horizontally to the rear to form a shelf on which the battery and other accessories are mounted. At the rear, this panel is curved upwards and spot welded to the scuttle, the bonnet surround being flanged and spot welded on the same joint. To give the dash shelf adequate stiffness for supporting the weight of the various components mounted on it, it is pressed to a form giving broad ribs parallel with the longitudinal axis of the car. On the left and right, this panel is spot welded to the dash side.

A mud baffle is positioned vertically in the wing above the front edge of the toe board, and bolted to the dash side. Its lower edge is spot welded to the toe board which, between the dash side and the wing, is extended and shaped to form part of the wheel-arch. The outer edge of the wheel-arch and baffle, where it bears against the wing, is flanged to the rear and sealed by a rubber strip inserted in a channel formed by spot welding a joggled strip to the front face.

A simple Z-section forms the front pillar. Its inner flange is spot welded to the dash side, while its outer flange carries the bolted-on wing panel. The incorporation of removable front wings is a noteworthy feature since it facilitates repair. Two U-shaped brackets support the front door hinges. The arms of the U are flanged on each side and tapered, the flanges nearest the centre of the car being spot welded to the dash side. Support for the front end of the lower bracket is provided by extending it forwards to a point close to the wheel-arch. The front end of the much shorter top bracket is supported by the nearly vertical edge of the dash shelf, where it is spot welded to the dash side. This arrangement of hinge brackets has made it possible to use an extremely light pillar structure while at the same time providing a rigid support for the door hinges, the countersunk bolts for which are carried in tapping plates held in a cage in the base of each U-bracket.

The structural function of the dash side is to carry the bending loads from the front end and transfer them into the body sill. It therefore acts as a cantilever beam, which has the top flange formed by the joint with the bonnet surround, and the bottom flange by its junction with the wheel-arch. At first sight, it might appear to be unsafe to use a light panel, of thin section, as a main structural member, but this practice is perfectly sound. Because of its great depth as compared with a normal chassis frame, the stresses induced in this beam are very light. Moreover, because they are completely stabilized by the bonnet surround and the wheel-arch, the flanges cannot fail by buckling.

The sill

It is a relatively simple matter to transfer the loads into the sill. To do this effectively, the sill inner plate is lapped over and spot welded to the dash, and beneath the pillar, its vertical front edge is flanged towards the inside of the section. The flange is thus positioned to continue downwards the local stiffening effect of the pillar in order that end loads from the pillar may be transmitted to the sill. Initial reaction to these loads is by shear in the inner plate, but they are rapidly diffused throughout the whole section. The sill outer panel is tapered from the pillar until it terminates at the front wheel-arch. Its flanges are spot welded to the dash side. The end load, due to bending, at the junction between the wheel-arch and the dash side is equal and opposite to that in the pillar, and the vertical shear is superimposed on the bending loads.

Although the centre pillar is intended primarily as a support for the rear door hinges and the front door latch, it also adds to the overall strength of the vehicle. The pillar is a simple box section formed by spot welding two top hat sections together, the inner section being very shallow. Considerable difficulty was experienced in designing an adequately strong structure where the rear end of the sill joins the rocker panel. In the Austin '7' the problem was even more acute than would have been the case in a larger model. This is because there are four doors, which of course take up a large proportion of such a small wheelbase. With a two-door saloon, it is possible to design a structure that is self-sufficient at the base without the assistance of the roof and windscreen pillars. In a four-door saloon, however, there is little doubt that these members play an appreciable part so far as the overall strength of the vehicle is concerned.

Heel board, wheel-arch and rocker panel

An inch or two from the rear end of the sill, the outer edges of the heel board are flanged forward, and spot welded to the inner plate. In the early stages of the design it was considered that additional reinforcement would be required towards the centre of the heel board where its strength is reduced considerably by the cut-out for the propeller shaft tunnel. However, subsequent tests have shown this reinforcement to be unnecessary, and it is no longer incorporated in the design. Because of the extreme stiffness of the structure between the bracket for the spring eye and the

sill, loads that would otherwise be transferred across this cut-out are transmitted cantilever-fashion into the sill, up the rocker panel, and finally across the rear seat squab.

The rear edge of the sill inner plate is flanged outwards, that is to say into the sill section, and the front of the wheel-arch is spot welded to it. Also spot welded to this joint is a plate shaped so as to close the rear end of the sill, and to prevent water and mud from being flung in off the wheel. The space between the front of the wheel-arch and the heel board is completely boxed in. The top is sealed by a 20 s.w.g. closing plate, which is spot welded on to form a continuation, around the front of the wheel-arch, of the seat pan. Two more closing plates are fitted. One of 20 s.w.g. is between the lower edge of the heel board and wheel-arch. The other is welded to the bracket carrying the spring eye. This plate is extended to the rear, where it is attached to and reinforces the wheel-arch, to give added strength for carrying the bolted-on shock absorber. The outer edge of the wheel-arch is spot welded to the inner edge of the rocker panel which, at the back, is extended upwards to form the D-post. The rear inner panel is spot welded to this post as well as to the top of the wheel-arch over which it is carried to the extreme rear of the car.

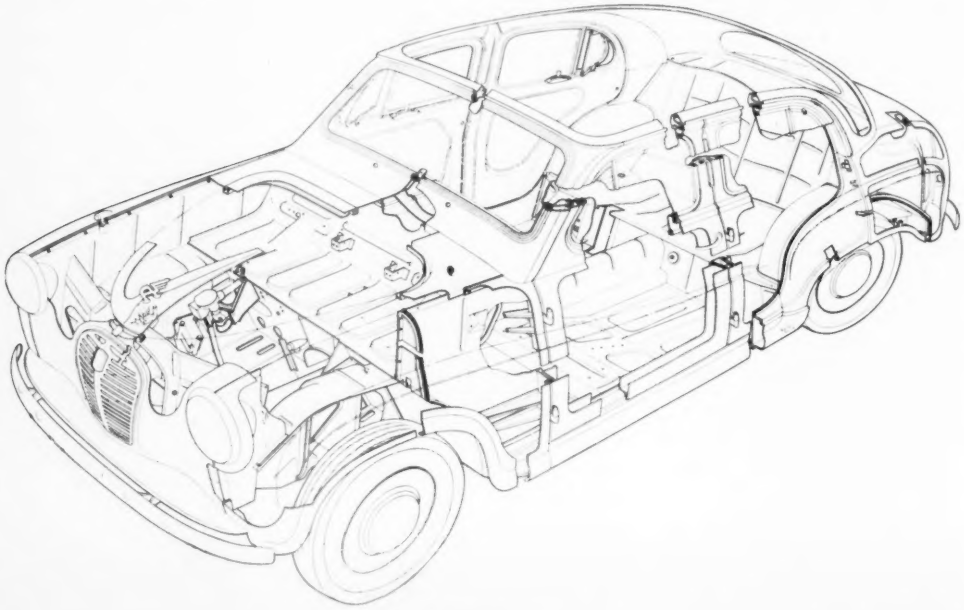
Stabilization of the wheel-arch to prevent undue deflection and local buckling is effected as follows. It is spot welded to the seat pan and to the boot floor at its lower edge, and at the extreme rear edge to the top of the back cross member. Additional stabilization is afforded by the side member, or inner quarter panel, and further forward, by the rocker panel. Although not essential for the rigidity of the wheel-arch, the rear seat squab is spot welded to it. The squab is also spot welded to the inner quarter panels and to the glove shelf, thus completely closing the rear end of the saloon body. Originally the squab was shaped so as to clear the telescopic shock absorbers which were fitted in the first prototype. Later, it was found easier to mount the lever-arm type, but the shaped squab was retained, as a feature that adds to comfort.

A horizontal solderless joint is incorporated at the junction between the sill and the rocker panel. This panel is spot welded to the vertical part of the shoulder, which extends round the door openings, in the orthodox manner, to form the outer door stop. At the rear, the rocker panel is similarly attached to the quarter panel. The outer door panel is extended down, completely covering the rocker panel to keep it clean, so that it does not soil the clothes of people entering or leaving the rear seats. To the rear of the rocker panel, the seal between the wheel-arch and the quarter panel is formed by a rubber strip carried in a channel formed, in the same way as at the front, by a spot welded-on joggled steel strip.

Rear end structure

Loads at the rear end are applied through the springs and, as at the front, reaction is provided by the inertia of the vehicle. The front spring eye is carried in a two-piece bracket spot welded to the heel board and to the rear floor. Both parts of the bracket are 18 s.w.g. The larger portion extends from top to bottom of the heel board. It is a top hat section, with the flanges spot welded to the heel board. The other portion of the bracket is a channel-section support, extending the whole length of the rear floor between the bottom of the foot ramp and the heel board; it is continued into, and spot welded to the sides of, the top hat section. Holes, for the pin carrying the spring eye above the support channel, are punched in the side of the top hat section. To allow the spring to be inserted from the rear, the base of the top hat section is cut away. In order to provide a large enough bearing area for the pin, further reinforcement is furnished by an 18 s.w.g. U-piece, spot welded to the sides of the vertical portion and to the top of the channel-section. All loads in the longitudinal, vertical plane are therefore split into two components. Reaction to the horizontal component is by shear in the floor, while the reaction to the nearly vertical one, taken by the top hat section bracket, is by the shear in the heel board. Side loads are transferred principally to the lower portion of the bracket, whence they are taken to the rear floor.

At the rear, the spring shackle is carried in a bracket which is fabricated in the following manner. In general, it is an inverted top hat section, the



The principal body sections

rear ends of its vertical side walls being flanged outwards and spot welded to a cross member, 3½ in deep. Its upper flanges are spot welded to the boot floor. The base is extended both to the front and to the rear. Its forward extension is bent upwards, flanged and spot welded to the boot floor, the edges being welded to the side pieces of the main top hat section. The rearward extension is spot welded under the rear of the boot floor. A step down in the boot floor forms the cross member that carries the loads from this bracket to the main side members, which are formed by the inner quarter panel. The upper edge of this panel is stabilized by its being spot welded to the water channel, down the side of the boot opening, and to the sides of the rear glove shelf.

From the cross member, the loads from the spring shackle, which are approximately half the total load applied at the wheels, are transferred through the rear side member and the vertical face of the wheel-arch, to the rocker panel and the heel board, whence they are transferred into the sill. A fair proportion of the load must also pass through the canopy, where it will be reacted by the centre pillar and by the windscreen pillar. Loads to be transferred forwards will be relieved by any weight placed, in the boot, on top of the rear cross member.

Other features

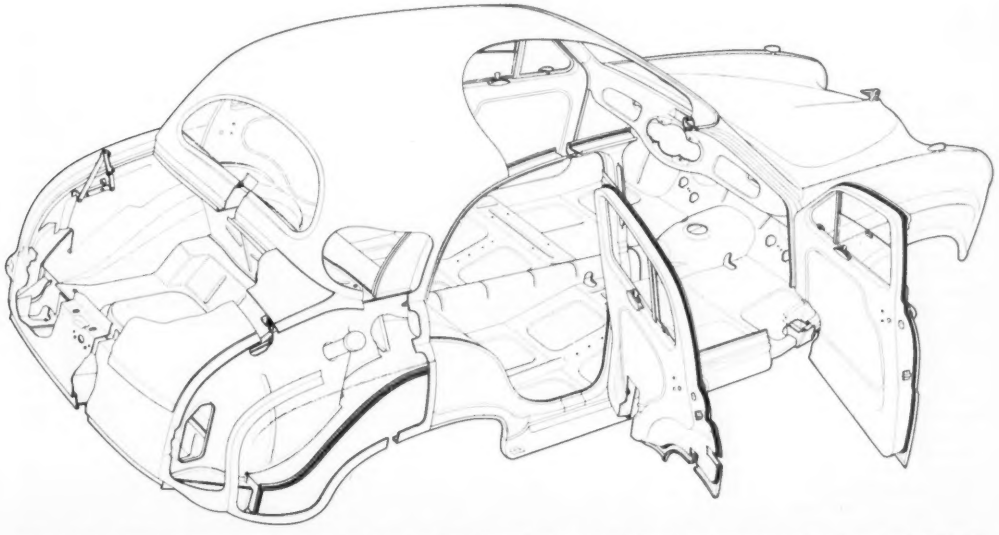
The main features of the structure have now been dealt with, but the mounting for two other major components, the front seats and the engine, are of interest. The front seats are each supported on a cross member which has its ends attached to the propeller shaft tunnel and the sill inner plate. This cross member, which is 3 in deep, is formed by a step in the front floor. At the top of the step, the floor extends horizontally about 4½ in to the rear, and it is then sloped downwards to the rear floor to form, beneath the front seats, a ramp on which the passengers in the back seats may place their feet. Two U-shaped tubes, of 1 in diameter, under each seat form the parallel link mechanism on which the adjustment is based. One of these tubes rests on the horizontal portion at the top of the step. Two steel straps bolted over it hold it in position. The upper ends of the arms of the U are pivot-mounted on lugs on each side of the forward edge of the seat pan. Behind this, the other U-shaped tube is inverted with its ends pivoted, one on the propeller shaft tunnel and the other on the sill inner plate. Carried on the horizontal part of this tube is the rear edge of the seat pan. This arrangement ensures that the floor is unobstructed beneath the seat.

The three-position adjustment is obtained in the following manner. A peg, projecting horizontally from the inner face of each of the side arms of the front tube, engages, in the unlocked position, in a horizontal slot in a plate welded on each side of another U-shaped tube. In the locked position, the pegs engage in notches cut in the upper edges of the slots, so that the weight of the tube keeps the notched plate in engagement with them. The ends of the tube that carries the plates are pivoted beneath the rear of the seat, one on the wheel-arch, and the other on the sill inner plate. Adjustment is effected by lifting the horizontal portion of this U-tube, which extends laterally underneath the front edge of the seat pan. This lifts the slotted plates, thus causing the pegs to disengage from the notches and enter the horizontal slot, so that the seat may be moved forward, or to the rear. There are, of course, three notches cut in each plate.

A channel-section is spot-welded round the inside of the propeller shaft tunnel to carry the rubber ring that forms the rear engine mounting. The ring is held in position by a pressed steel bridge-piece bolted across the lower portion of the tunnel. Thus the tunnel, supported at the rear by the heel board, and at the front by the toe board, acts as a beam to carry the rear engine mounting, and half of the front seat loads. The other half of these seat loads are taken directly by the sill.

A simple bracket forms an inclined platform, on which the sandwich type front engine mounting is carried. To the front and rear of the platform, the 14 s.w.g. sheet, of which the bracket is made, is bent down and, at the lower edges, is flanged outwards and drilled for the four ⅜ in diameter bolts that secure it to the top of the side frame. The weight of the engine is taken, therefore, through the bracket, and supported partly by the vertical inner wall of the wheel-arch which is spot welded to the side frame, and partly in bending by the inner flange of the frame and the L-section reinforcing plate, spot welded to it.

There is one feature of the design, which at first sight may appear to be a minor detail, but is in fact of great importance. This, to use the



A bracket and a plate reinforcement for slinging the body are fitted respectively under the front floor ramp and under the rear end of the sill on each side

The rear brake cylinder is mounted adjacent to the spring anchorage bracket on the heel board

terminology of the Austin Motor Co., is the "'swaging." It must be emphasized that in this context the term "swaging" has a meaning very different from the normal engineering usage. Here it means that one or more flutes have been pressed into the panel. In what follows, such a flute is referred to as a swage.

Swaging may be adopted for two reasons. It can be used to stiffen the panels against vibration, thereby reducing noise, or it may be adopted to fulfil a structural function, either as a strut in a panel that takes compressive loads or for stiffening a panel forming the web of a beam. A flat panel is incapable of taking any appreciable loads in compression. If, however, it has a channel-section swage pressed into it in the appropriate position, the swage will act as a strut, and thereby increase the compressive load bearing capabilities of the panel.

Some examples of the use, in this car, of swages as struts are as follows. Where the outer edges of the side members of the portal frame are swept in to clear the front coil springs, there is a swage positioned transversely between the two flanged edges of each member. This swage supports the curved edge when it is loaded compressively. Under these conditions, if the swage is not there, the radius of curvature would tend to decrease, and the edge would move inwards, buckling the flat surface of the member. A swage or any other form of strut used in this way is usually termed a *kink strut*. These struts should, without fail, be incorporated wherever a beam of this type is kinked.

Another example of the use of a swage as a strut is in the rear quarter inner panel. This is swaged vertically immediately above the rear cross member, the loads from which are passed directly into it. Although the upper portion of this swaged panel is curved, it is probable that the compressive loads imparted to its lower end are diffused into the panel in shear, so that very little compression is taken by the top end.

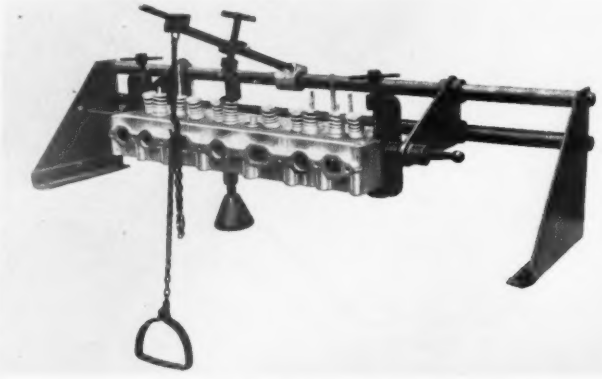
When used to stiffen panels that form webs of beams, swages prevent the formation, under normal loads, of a tension field. The principles governing the use of swages for this purpose are too involved to be described here, but they may be found in any standard textbook in which Wagner beams are described. One of the best examples of this type of swaging is to be found on the front side member, between the dash and the radiator bulkhead. Others are to be found on the rear seat squab and the parcel shelf.

CYLINDER HEAD DECARBONIZING JIG

AN oft-quoted advantage of the overhead valve type engine is that the cylinder head can be lifted and transferred to the bench for decarbonizing and valve grinding operations. The full benefit of this feature can only be realized, especially in the case of the larger engines, if the head can be conveniently positioned and handled on the bench. The HF jig to facilitate this work has now been redesigned and enlarged to accommodate heads from the largest engines of commercial vehicles.

Parallel slide bars mounted on end frames secured to the bench carry two brackets furnished with rotatable clamps. The cylinder head, held rigidly in the clamps, can be turned through 360 deg and locked in any desired position by the lever on the left-hand clamp. To remove the valves, the head is locked at the position determined by the angle of the valves, and an adjustable dolly placed to support the valve head. The spring compressing arm, pivoted on a third sliding member, has a reversible forked end to deal with different type retainers and is operated by a chain and stirrup. Thus the operator is left with both hands free to remove the cotters.

For other operations such as decarbonizing, valve seat trueing and grinding-in valves, the head is swung to a convenient position and, obviously, the disposition of inlet and exhaust valves at different angles presents no difficulty. Cylinder heads to maximum dimensions of 47 in long, 14 in wide and 5½ in deep can be handled on this equipment. The manufacturers are Harvey Frost and Co. Ltd. of Bishop's Stortford, Herts. (2024)



The HF decarbonizing jig

POWER STEERING

A Survey of American Systems for Private Cars

IN a product which has undergone such intensive development as the modern American automobile, it is a striking anomaly that, in one respect, it is definitely inferior to examples dating back twenty or thirty years. Greatly increased front-end weight, together with the use of tyres much larger in section and inflated to a lower pressure, make the steering of the modern car distinctly unsatisfactory.

The low-geared mechanisms, while keeping parking effort at the rim of the steering wheel down to accepted amounts, involve as many as 5·5 turns from lock to lock in the case of a heavy and refined vehicle. Handling on the open road is correspondingly sluggish and at low speeds tends, among other things, to bad urban traffic manners. Drivers may take so long to apply lock that they tend to pull out of their traffic lane into the centre of the road when making a right-angled turn.

In an emergency at speed, a low-geared steering can be actually dangerous from slow response, the effect being much aggravated by the big slip angles of modern tyres, which demand far greater overall movement when negotiating anything of the nature of a serpentine course.

The steering of the higher-priced modern American car is all the more unsatisfactory by contrast with the rest of the vehicle. Duo-Servo hydraulic brakes are relatively effortless, automatic transmissions eliminate the clutch pedal, while windows and coupé hoods are increasingly provided with power operation at the flick of a switch.

No blame attaches to American engineers for this state of affairs: the high front-end weights are an accepted advantage in respect of suspension and passenger comfort and the low-pressure tyre, which is the principal culprit, would appear to be an essential feature of the modern automobile.

Unfavourable changes in load distribution and the use of tyres of increasingly large section and greater parking resistance have been met, as far as possible, by improvements in the efficiency of the manual steering gear. Of this a good picture is given in a brochure issued by the Saginaw Steering Division of General Motors. To emphasize the point two extremes are taken—the 1929 Chevrolet and the 1951 Cadillac. The Chevrolet had a kerb weight of 2,672 pounds, of which only 1,219 pounds was carried by the high-pressure front tyres. A worm-and-sector steering gear of 9·5 : 1 ratio was adequate to keep parking effort down to an accepted standard of 45 lb at the rim of the steering wheel, while about 2·5 turns sufficed to go from lock to lock.

As, in the course of the intervening years, the demands on the steering mechanism increased, successive improvements were made in design. The first step was the change from globoid worm and two-toothed sector to what may be described as a version of the Marles gear, with a single roller tooth running on a plain bush and backed up by parallel-roller thrust washers.

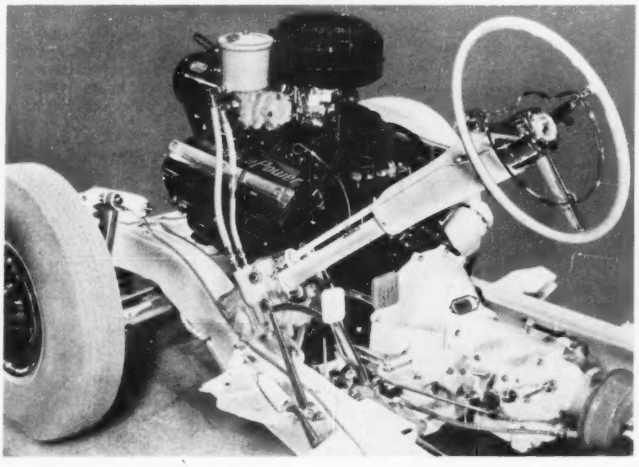
A further improvement in efficiency, both forward and reverse, came with the adoption of a two-toothed roller mounted on integral angular-contact ball bearings, while a final improvement in efficiency of about 10 per cent resulted from the employment of a gear of the recirculating ball type, the nut operating the pitman shaft through rack and sector.

Nevertheless, the 1951 Cadillac has to use a box ratio of 21·3 : 1, or 5·3 turns to the lock, to keep the parking effort down to the same figure of 45 lb at the steering-wheel rim which was the accepted standard thirty years ago.

The kerb weight of the large car is 4,290 lb and the front end weight is 2,250 lb, while the tyres are of larger section, smaller diameter, and work at lower pressures.

While power steering was clearly indicated, its cost was for long felt to be an insuperable handicap. Intensive development work has now made it possible to offer this refinement at an approximate present-day extra cost of 200 dollars to the public. First featured by Chrysler in 1951, it was available on the more expensive models of five firms in 1952. Public response has been even more enthusiastic than was expected and demand exceeds supply at the moment.

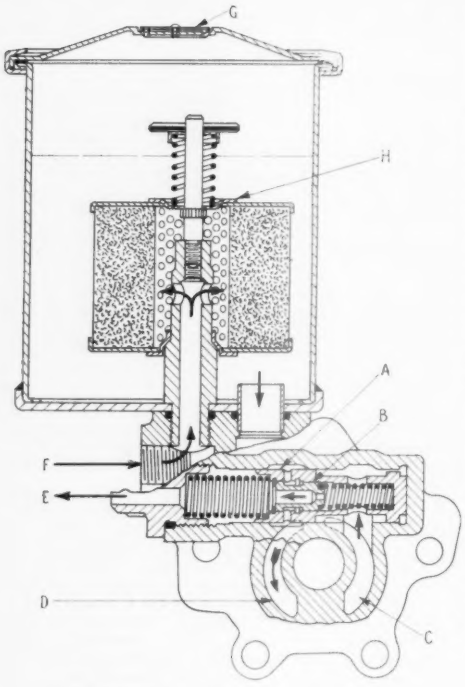
The innovation has not been without its teething troubles in the development stages, the most serious being the noises set up by pulsations in the oil flow and transmitted into the passenger compartment by way of the steering box and frame or by the steering column. Detail modifications in pump design and manufacture and the use of extensible rubber hoses have helped to reduce this disadvantage but it is, in general, still apparent when parking. The oil pressures reach 600 to 800 lb/in^2 under this condition and the pumps tend to be rather noisy.



Installation on Chrysler vehicle

Even in this early stage of development some degree of standardization of practice is appearing, though some firms show caution in taking full advantage of the possibilities. General Motors, up to the time of writing, do not alter the steering box ratio when supplementing it by power application. Full lock still requires 5·3 turns, but the effort is enormously reduced; maximum parking effort at the steering wheel rim is under 10 lb, as compared with 45 lb for the unassisted steering.

Chrysler, however, have changed to a box ration taking 3·5 turns to the lock as against 5·3 turns with the original manual box. Although quicker steering is greatly appreciated by



A Flow control valve (closed)
B Pressure relief valve (closed)
C Rotor discharge
D Rotor suction
E Oil to valve block
F Oil from valve block
G Vent valve
H Filter relief valve

Eaton pump and flow control valve

experienced drivers, others take a little time to acquire familiarity. It would seem, however, that the quicker steering will soon be standardized by all makers. It must be admitted that the power assistance does, in practice, make the steering more responsive, since the driver has not to take a grip and assume an attitude demanded by the exertion of considerable torque.

All makers arrange, by a check valve on the oil supply system, that the gear can be operated manually in case of stoppage of the engine or breakage of the belt or belts driving the oil pump. High efficiency of the manual gear is therefore still taken as essential; the Saginaw-General Motors gear, for instance, still uses the recirculating ball mechanism.

All makers agree in providing preloaded centring springs in the floating part which constitutes or operates the power control spool valve. These are generally set to a steering rim load of about 4 lb or about 8 per cent of assumed maximum manual effort with an unassisted gear. Below this figure, which covers a good deal of steering at speed on a smooth road, the action is directly manual and without the backlash which becomes evident when valve travel commences.

The front wheels can straighten themselves out after a curve without causing the power application to operate. At the same time, advantage has been taken of the resistance to undesired reverse movement held in reserve by the power mechanism to allow, in some cases, much more castor than formerly, since an additional two pounds or so on the wheel rim will hold the car on a curve even when a large castor angle is employed.

Two out of the three gears to be described give definite power "feel," that is to say, the control is strictly a servo one, increased power assistance demanding a corresponding slight increase in manual torque.

The extent of this servo action was the subject of many comparative tests by the Chrysler Company with cars in the hands of drivers to whom power steering was a complete novelty. On the first run many drivers preferred a 50 per cent "feel," that is to say, a setting in which the engine was only doing half the work. After a few minutes' experience, however, they became quite at home with the finally standardized setting of 12 per cent (engine doing 88 per cent of the work) and thereafter greatly preferred it.

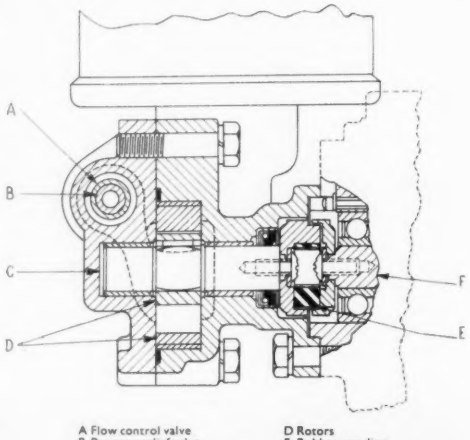
More - or - less general agreement seems to have been reached as to the maximum steering speed given by the pump when parking a car with the engine ticking over. Something between 50 and 60 r.p.m. of the steering wheel is normally arranged for. It has been found that anything less than this causes a most unpleasant feeling to the driver if he can "run ahead" of the power and lose its assistance.

With a heavy car it is clearly inadvisable to operate without power assistance on the highway; coasting with engine at rest cannot, therefore, be indulged in. This is of no importance to American users, to whom a slight fuel economy is a small matter.

The various forms of gear-type pumps run at about engine speed and circulate in the pump body about 1·65 gal/min on a fast tick-over and 6·66 to 8·3 gal/min at the high cruising speeds common in the United States. In all cases a flow control valve is fitted which allows most of the oil to return direct from the pump delivery port to the suction port when the engine is running fast. This is a most essential feature since, to give quick servo response, the spool valve travels are very small and the port areas much restricted. Were the whole pump delivery to be forced through the ports in the neutral position and at high pump speeds, a heavy back-pressure would be set up, causing appreciable power absorption and rise in oil temperature.

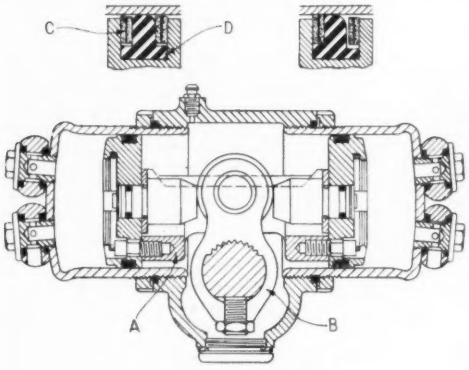
Largely on account of these flow control valves, the pump losses when the control valve is in neutral are not serious. On the tick-over they are not capable of measurement but are of the order of $\frac{1}{4}$ h.p. at 60 m.p.h. The value of the control valve is clearly demonstrated by curves given by the Chrysler Corporation, which show that, if all the pump delivery at about 60 m.p.h. were actually circulated through the servo spool valve ports in neutral, the power absorbed would be over 2 h.p.

Power required to operate the steering gear is no serious matter, being of the order of 1·5 h.p. maximum on the tick-over when parking on a rough concrete surface. When the vehicle is moving fast, full steering power is never required and a maximum of



A Flow control valve
B Pressure relief valve
C Thrust washer
D Rotors
E Rubber coupling
F Dynamo shaft

Eaton oil pump drive



A Yoke
B Power arm
C Two phenolic rings
D Rubber T-ring

Power cylinders, with details of piston packing

about 2 h.p., and that momentary, is all that need be provided.

At the same time we imagine that some effect on the petrol consumption may be felt from the necessity to give a fairly fast and robust tick-over setting, if the engine is not to be accidentally stalled when parking. On a car with automatic transmission not in neutral, any acceleration of the engine to give parking power might lead to undesired movement of the vehicle.

In any case the increase in fuel consumption caused by the adoption of power steering should be far less than that already accepted when an automatic transmission is used; fuel economy is, in any case, not so important in America as in Britain.

Chrysler power steering

This was the first power steering to be offered to the public on a pleasure car; it is also the most elaborate and refined, the makers setting themselves a very high standard. Much of our information on this equipment comes from a paper by W. A. Hunter, Project Engineer of the Chrysler Corporation, which is refreshingly frank and outspoken in respect of troubles encountered in development and is correspondingly valuable to the automobile engineer making his first approach to the problem.

If we abstract a considerable amount from this paper readers will understand that no reflection on the other makers is intended. They have tackled the problems in an equally painstaking, if sometimes simpler, manner and references to the Chrysler must be taken as applying more or less to the other gears to be described.

Oil pump

The Chrysler engineers started with the assumption that the gear should be capable of rotating the steering wheel at 60 r.p.m. with the engine and pump running at 800 r.p.m. The maximum working oil pressure was set at 650 lb/in^2 and, with the operating pistons appropriate to this pressure, an oil flow of 1·25 gal/min was called for. Oil pump dimensions are such that this delivery is maintained at 800 r.p.m. even when the volumetric efficiency of the pump has fallen by wear in service to 50 per cent.

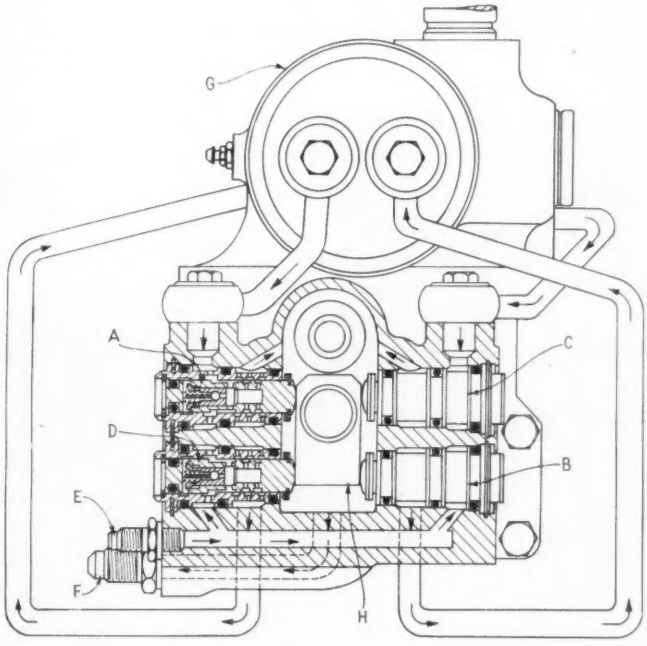
The Gerotor pump is carried by the dynamo end plate and is driven by a rubber-cushioned claw coupling. It takes its supply from a container carried directly by the pump body and having a capacity of three pints, and S.A.E. 10 W engine oil is recommended. Chrysler do not employ an automatic transmission of the torque-converter type; on other makes of vehicle so equipped the transmission oil is specified for the power steering also.

After passing through a flow control valve, to be described, the regulated quantity of oil necessary for the gear operation passes to the valve block on the steering box assembly by a rubber hose and returns in similar manner to the oil tank after passing through a cartridge filter fitted with a filter relief valve set to about 6 lb/in^2. The air space above the fluid is normally sealed, but a spring-loaded vent valve opens on a rise of about 5 lb/in^2.

The filter equipment is a vital feature of all the power steering gears, since the piston control valves work with very fine clearances and could easily be jammed by dirt particles. Clean oil is also essential for the oil pumps. Though only loaded to the full for extremely short periods, they are, in all cases, working at distinctly high pressures for the various types employed.

Flow control valve

All the gears to be described use some form of control valve to limit the normal delivery to the control valve to from 1·25 gal/min (Chrysler) to 2·1 gal/min (Ross). The Chrysler device is combined with the relief valve and consists of a sliding member lightly end-loaded by a spring and machined to take an inner piston valve loaded by a strong spring to act as relief valve at about 650 lb/in^2. Delivery from the rotor discharge port passes through radial holes in the flow control valve and then axially through



A Left-turn reaction valve
B Left-turn distribution valve
C Right-turn reaction valve
D Right-turn distribution valve
E From pump
F To reservoir
G Power cylinder
H Valve operating block

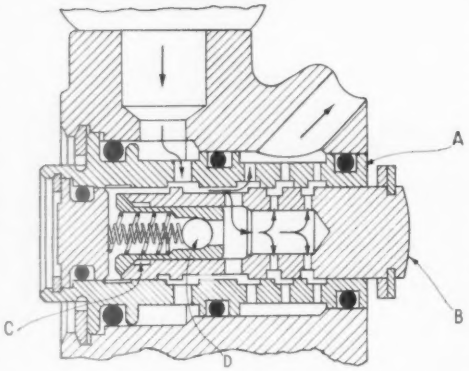
Valve body assembly, valve in neutral position

a metering orifice drilled in the relief valve piston.

Above 1·25 gal/min the resistance offered by the metering hole is such that a back pressure is set up sufficient to overcome the control valve spring and move the valve so that a land on it uncovers an annular port in the casting, allowing the excess delivery to pass outside the control valve body back to the rotor suction port.

Since the pump, at high engine speeds, delivers about 8·3 gal/min it will be clear that the flow control valve, in by-passing most of this reduces the back pressure set up across the restricted passages of the steering control valve to a most important extent — back pressure being roughly proportional to the square of the fluid velocity. The pressure relief valve is of the piston type and moves to the right, allowing the oil to escape by radial holes in the flow control valve body back to the rotor suction port.

By far the most difficult part of the development work by the Chrysler engineers was the elimination of noises set up by the slightly pulsating flow of the gear pump. These noises were transmitted up the steering column and from the steering box to the frame and hence to the body structure.



A Valve sleeve
B Valve
C Dashpot orifice
D Check valve

Valve detail

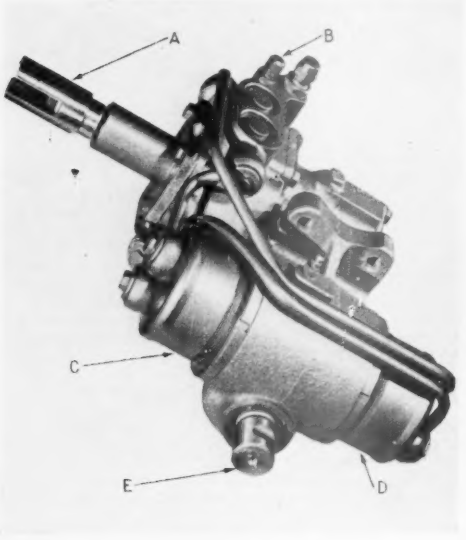
That they were in no sense airborne was proved by the fact that they could be eliminated by mounting the steering box in soft rubber and by interposing rubber cushions in the steering mast and column. While the last two were retained, to a reduced extent, in the production models, rubber mounting of the steering box of a softness adequate to stop the noise led to dangerous steering instability.

Recourse was at first had to a duplex pump, with two rotors timed half a tooth space apart, but even with this results were not considered satisfactory. The most valuable feature was found to be the inclusion of a length of special $\frac{3}{8}$ in highly expansible hose in series with a standard $\frac{1}{4}$ in hose, this being placed in the delivery line. Similar arrangements in the return line were ineffective. Modification of rotor tooth form effected considerable improvement and changes in the location of suction and discharge port edges reduced cavitation and "trapped oil" noises.

A notable reduction in noise was made by taking care that the return oil from the flow control valve blends smoothly with the flow from the reservoir. The original T-junction set up a vacuum of several pounds per square inch at high speeds by turbulence, causing cavitation in the pump rotor. In the final arrangement a linear delivery curve was obtained up to 5,000 r.p.m. pump speed—a remarkable achievement in a gear-type pump. Air in the oil was another source of noise and it was found necessary to return the flow from the steering box to the reservoir instead of direct to the pump, using the reservoir as "make-up" only.

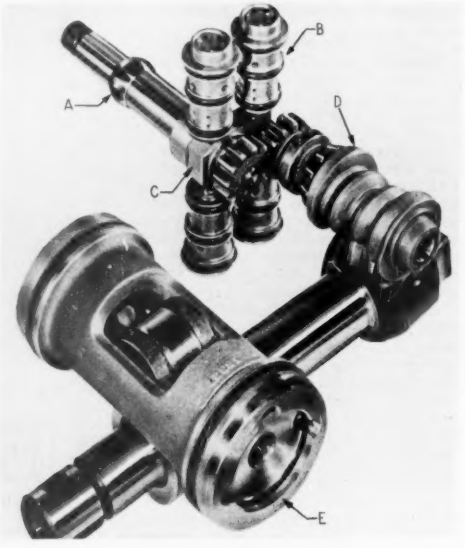
Power steering unit

The manual steering gear is of the worm and roller type, with the worm mounted on taper roller bearings.



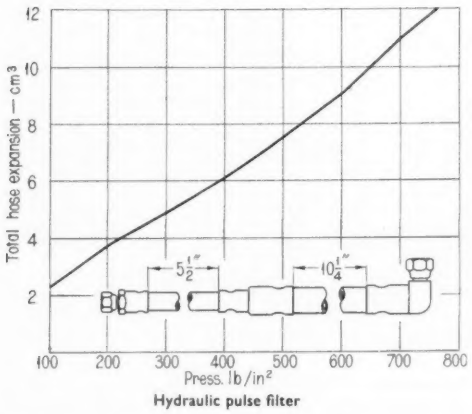
A Steering column coupling
B Valve body assembly
C Left-turn cylinder
D Right-turn cylinder
E Steering gear shaft

Chrysler power steering unit

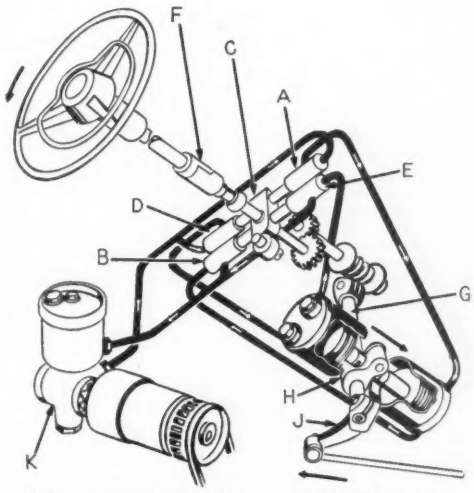


A Spherical bearing
B Valves
C Valve block
D Worm and roller
E Piston assembly

Internal components of Chrysler unit



Hydraulic pulse filter



A Right-turn distribution valve (closed)
B Right-turn reaction valve (open)
C Valve block (up position)
D Left-turn distribution valve (open)
E Left-turn reaction valve (closed)
F Flexible rubber coupling
G Steering shaft
H Power arm
J Drop arm
K Oil pump

Flow diagram at left turn on Chrysler equipment

Parallel to it comes the power piston assembly, comprising two opposed 3 in diameter pistons acting on a roller carried in needle bearings in an upstanding arm splined to the drop-arm shaft.

A notable feature is the sealing of the pistons by T-section rubber rings, each fitted between two pairs of piston rings of phenolic material. The rubber ring is in light contact with the pressed-steel cylinder and the phenolic rings are normally clear. When pressure is applied it is taken by the rubber, which swells within the down-stream piston rings, forcing them into contact with the cylinder and preventing extrusion of the rubber past them although, in the unloaded condition, they have appreciable clearance from the cylinder.

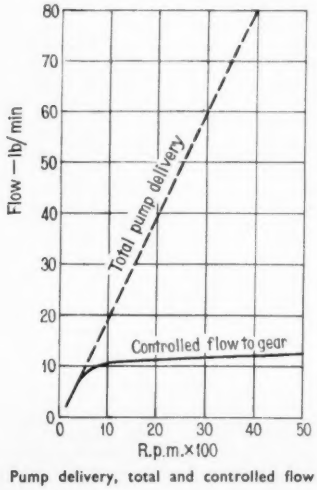
The object of this ingenious arrangement is to reduce to the absolute minimum the friction of the gear when power is not applied, so that light and small steering movements can be made by direct manual effort and the gear is free for self-centring.

The control valve mechanism is highly refined and ingenious. A short, spherically-mounted extension shaft is coupled to the steering column shaft by a rubber-lined universal joint. Its lower end carries, rotatably, the valve operating block which is located between two pairs of opposed piston valves, lightly spring-loaded into contact with it. Below this comes a spur gear, engaging a similar one on the worm shaft of the manual gear, and kept closely in mesh by an eccentrically adjustable roller contacting one end of the valve operating block. Any torque applied to either the steering wheel or the worm shaft causes the spur gear on the extension shaft to climb up or down the spur on the worm shaft, moving the operating block and the valves.

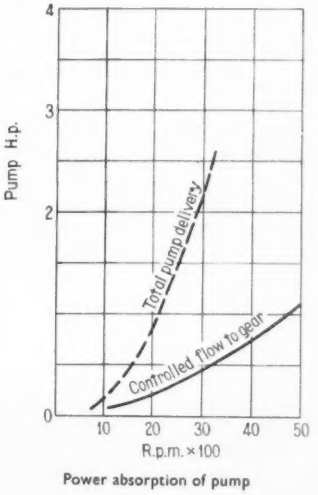
All four valves are identical in construction. They are only 0·57 in diameter, but each has triple lands acting in parallel and giving a fairly large area for a very small travel. Some distance from the axis of the extension shaft come the opposed pair of distribution valves, each controlling the admission of oil from the pump to one end of the power cylinder. The other two, termed reaction valves, control the escape of oil from the power cylinder and are cross-connected by separate steel pipes.

The outer ends of the distribution valves are both permanently connected to the pump delivery line and are, therefore, always in hydraulic balance. The outer end of each reaction valve is in communication with its end of the power cylinder. Starting from neutral, with all valves partly open, a steering movement will close one distribution valve and open the other, full travel occurring here since there is no hydraulic unbalance to hinder it. As soon as the reaction valves begin to move, however, a difference of pressure is set up between the two ends of the power cylinder, causing a similar unbalance in the loads on the valves and giving a reaction, felt at the steering wheel, proportional to the operating pressure in the power cylinder.

The valve block is eccentrically mounted on the extension shaft in such a manner that the distribution valves are given full travel for a very small movement; steering control is then purely a function of the partial closing of one reaction valve. In all other power steering gears, one valve piston or spool only is employed and steering control is a function of the partial closing of a leaking supply port together with the partial closing of the escape port from the other end of the power cylinder.



Pump delivery, total and controlled flow

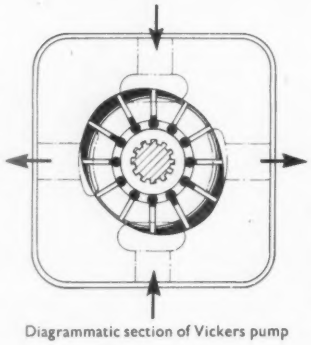


Power absorption of pump

The Chrysler system, though complicated, would seem to give more sensitive and responsive control. Like other and simpler systems, it gives proportional reaction on the steering wheel, the extension shaft leverages and the area of the reaction valves being chosen to give a manual torque on the drop-arm shaft of 1·73 P, a power torque of 12·30 P and a total output of 14·03 P, where P is the hydraulic pressure existing at the moment.

An important feature of the valve arrangement is the partial blinding, subject to a check valve and a dashpot bleed orifice, of the outer ends of the valve pistons. This was found to prevent chattering of the system on rough roads at low speeds. It is interesting to note that the earlier models had single-ported valves with longer travel. Considerable valve hiss was audible and this was much reduced by the triple-ported valves. It would appear that a long thin stream of oil under pressure makes less noise than a shorter and thicker one.

As already mentioned, Chrysler go 3·5 turns of the steering wheel to the lock when power steering is fitted, compared with 5·5 turns on the original manual gear. It is worth noting that they intend to reduce the number of turns still further in years to come, as the public become accustomed to the advantages of quicker steering. We think that anyone catering for the English market could safely start at say 2·75 turns to the lock.



Diagrammatic section of Vickers pump

Saginaw power steering

This mechanism, which is made by a subsidiary of General Motors, is available on the larger models built by that concern. The particular gear described is fitted to the larger Buicks as an extra. By comparison with the Chrysler gear, the Saginaw is much simpler in many details, especially in the steering control valve. No noise insulation of the undivided steering column has been attempted and a certain amount of oil pump noise, referred to in the instruction book as "normal noise level" is accepted as inevitable. It is also interesting to note that the firm has, for the present, made no change in steering box ratio, which stays at 5·3 turns to the lock.

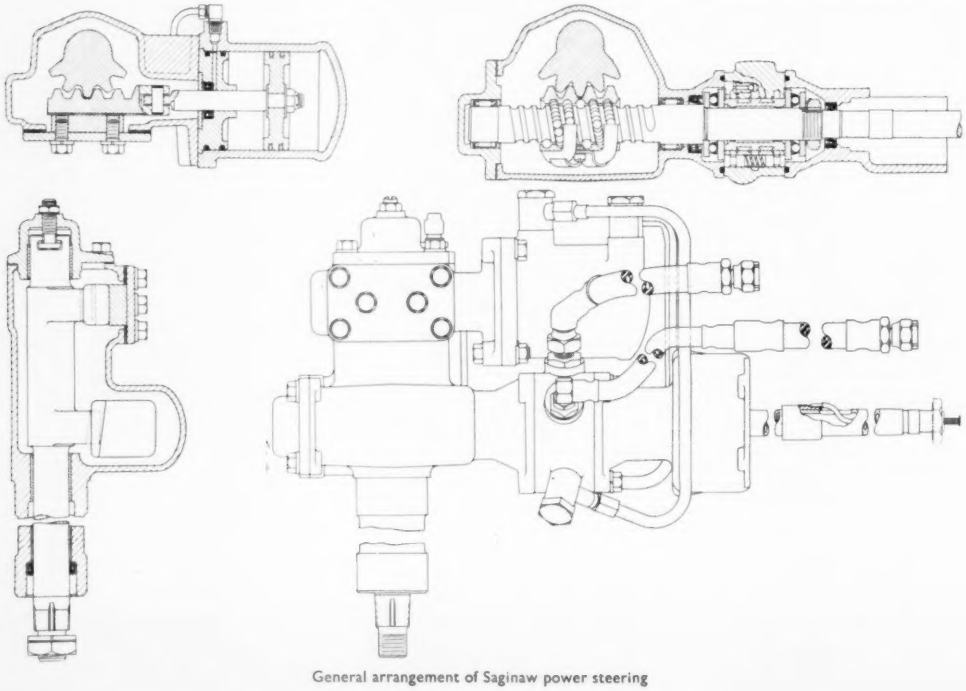
Oil pump

To facilitate supply, two alternative proprietary components are fitted, the Eaton Gerotor-type pump and the Vickers balanced-vane pump. Both are arranged to deliver about 1·25 gal/min on the tick-over; they have separate piston-type flow control valves embodied in the pump bodies and separate relief valves. The check valve which permits manual operation in the event of the pump being out of action is mounted in the body of the steering control valve, which is part of the steering box assembly.

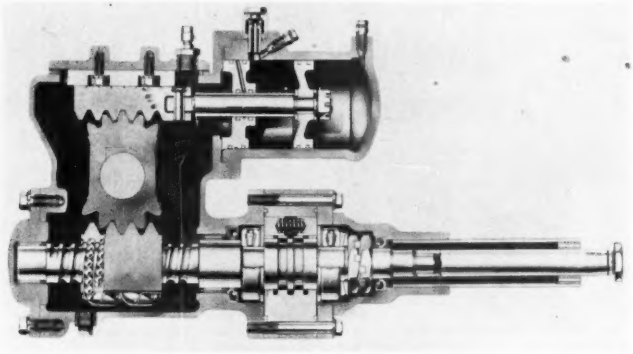
Flexible rubber hoses connect the oil pump to the control valve, and lighter flow and return hoses couple the pump with the oil reservoir which is attached to the cylinder head. These hoses, it will be understood, handle only the gear-operating quantity, which is kept down to 1·25 gal/min by the flow-control valve. This oil passes through the usual full-flow oil filter with by-pass valve.

Both Eaton and Vickers pumps have individual drives by a single V-belt; rated delivery is given at 450 r.p.m. of engine, at which speed the locked pressure should be 700 lb/in^2 at 170 deg Fahrenheit oil temperature. The relief valve is set at 750 lb/in^2.

The Eaton internal geared pump will be familiar to readers of the *Automobile Engineer*, but to those who do not recall the description of the Vickers pump given in the *Automobile Engineer*, November 1947, a few particulars of this unit may be of interest.



General arrangement of Saginaw power steering



Sectioned model of Saginaw steering unit

The cylindrical rotor is slotted to take twelve radially sliding vanes which are moved outwards, by centrifugal force plus oil pressure on their inner ends, against the pump ring. This is machined with two large and two smaller cylindrical portions joined by cam faces. There are two suction and two delivery ports to the two pumping chambers and the design gives a hydraulic balance to the rotor, which imposes no load on its bearings. An interesting feature is the taking up of end clearance by a floating end cover subject to fluid pressure, plus a weak spring to give an initial seal.

Steering box assembly

The basis of this is the recirculating-ball, manual gear already standard on the Buick. The screw has a single-start ball groove and the nut has two separate sets of 30 balls, one at each end of the nut, with the usual recirculating tubes. Thread pitch is about 0·41 in and ball diameter 0·28125 in.

Rack teeth cut on the nut engage a sector integral with the drop arm shaft; the teeth are wedge-shaped and clearance adjustment is by lateral displacement of the shaft, which gets its end-location from a cheese-headed, hardened setscrew, fitting in a T-slot cross-broached in the end of the drop arm shaft, with a selected shim-ring behind the head to give a close running fit.

The steering screw is carried on Torrington needle-roller bearings and is extended through an oil seal into the valve chamber. Here a hardened spool piston valve, of about 1·625 in diameter and having three lands, works in a body casting having a central annular chamber fed by the oil pump. Drillings on each side communicate by way of steel pipes with the two ends of the power cylinder, and end chambers are both connected to the oil return pipe.

The spool valve comes between two ball-thrust washers, adjusted to a close running fit between a shoulder on the screw shaft and a locknut secured by a tab washer. The valve body is 0·060 in shorter, measured over its end recesses, than the distance between the inner thrust rings, that is, so much shorter than the spool valve itself.

A maximum end travel of 0·030 in on either side of the central position, therefore, is possible. The larger diameter of the valve body is full length and has longitudinal holes in which slide pairs of plungers separated by stiff compression springs. The outer ends of these plungers bear, for a part of their diameter, on the flange faces of the steering box and column mast, while the remainder of the surface bears on the inner faces of the thrust rings.

The plungers and springs thus act as a pre-loading device tending to hold the spool valve in its central position, with all the ports partly open and the oil returning to the pump without pressure building up in either end of the power cylinder.

The springs are set to give a total load of 300 lb and until this end force on the screw and nut is exceeded no movement of the valve and no power application occurs. Beyond this value, representing about 3 lb on the rim of the steering wheel, the centring springs yield and the valve moves to shut off the oil supply to one end of the power cylinder and the oil escape from the other. A pressure is thus built up, regulated by the amount by which the escape is obstructed.

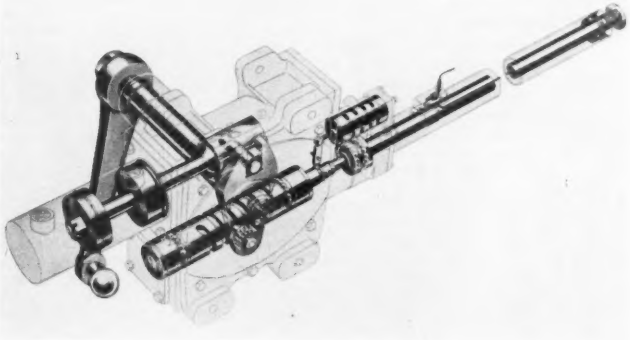
This build-up of pressure is applied to the spaces occupied by the compression springs and adds to the separating force they exert, and hence to the self-centring force exerted on the spool valve. The driver can thus "feel" the power applied and the diameter of the five sets of plungers is so chosen that about 70 per cent of the work is done by the power cylinder, the driver supplying the remaining 30 per cent. When the public becomes accustomed to power steering it is quite easy for the makers to increase the proportion of power applied by reducing the diameter or number of the centring plungers.

The screw shaft is continued to form the integral steering column which, with the steering wheel, partakes of the end movement of the control valve. In this respect the Saginaw gear differs from the Ross design. In that gear the column is separate, is given end location by a ball bearing, and is freely splined to the worm shaft, so that the steering wheel does not move up and down as the valve operates.

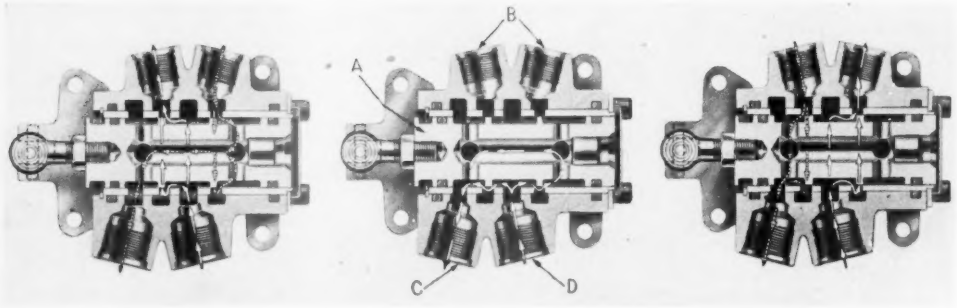
It should be noted that a full travel of the control valve from its neutral position corresponds to about 4½ in movement of the rim of the steering wheel, or 9 in movement from hard right to hard left. At first sight this may seem an excessive amount of backlash, but two factors must be borne in mind.

In the first place, for small steering movements on the open road at speed, 3 lb manual effort at the wheel rim is adequate without calling up the power. The centring springs, therefore, do not yield, the valve is held still, and there is no valve-motion backlash at all.

Secondly, the steering effort at speed is far less than when parking. Nothing approaching full pump pressure is called for and it is not even necessary actually to close the escape ports, let alone mask them by several thousandths of an inch, which occurs under full travel when parking. Actual "backlash" movements on the open road are therefore generally less than half the figures given.



Arrangement of Ross Hydrapower unit



Valve operation on Ross Hydrapower unit

Power cylinder

Slightly over 3 in diameter, the cast-iron power cylinder has a piston packed by two cast-iron spring rings. The piston rod passes through a close fit in the cylinder cover, between which and the oil seal is a vent led back to the return pipe. Any slight leakage is handled by this pipe, while the seal is relieved of power pressure. The cover is a piston fit in the bore and is sealed by two "O" rings of synthetic rubber.

Connection to the drop arm shaft is by a rack, pin-jointed to the piston rod and engaging a normally-toothed sector integral with the shaft. The rack is backed up by a pad secured to a cover, shims under which can be changed to give the correct tooth clearance.

Owing to the mass of the piston and the inertia damping of the oil in the steel pipes there is a tendency for a hammer and anvil action between sector and rack. Clearances must therefore be held fine at this point if rattle on the road is to be avoided.

A much larger version of this gear is being made for use on trucks. The power sector is in this case arranged directly above the manual sector, heavy taper-roller thrust washers are used on either side of the valve spool and other parts are more robust.

Ross power steering

The Ross Gear and Tool Company has been in moderate-scale production of hydraulic power steering for some considerable time; its efforts, however, have mainly been directed to commercial and military vehicles requiring larger mechanisms and greater drop arm torque than a pleasure car.

The P720 Ross gear approximates in layout to the larger Saginaw unit, the power cylinder coming above the manual gear and being coupled to the drop arm shaft by a sliding block, loosely pinned to the piston rod and having a vertical recess milled in it. In this recess works a roller carried by a stud projecting inwards at the upper end of a double-armed lever.

A peg, mounted on taper roller bearings, is used in the cam-type manual gear. At its lower end the cam is extended through a spool valve, with ball thrust washers and centring plungers identical in principle with those described in the Saginaw gear, but somewhat more easy of access since the mechanism comes at the free end of the shaft.

The steering column is integral with the cam and rotates and slides in a needle roller bearing at the top of the mast. It should be mentioned that the spool-type control valve co-axial with the steering column assembly was originated some years ago by the Bendix concern, and is known as the Bendix valve in America.

Though of larger dimensions, and intended for commercial vehicles, the Ross HP70 gear has special features which will be of interest to car designers. Turning on plain bearings on its largest diameter, the cam works with a roller-bearing peg as before, and obtains its end-location in the box casting from ball thrust bearings backed up by distance-pieces containing a number of compression springs preloaded by a coaxial plate retained in place by recessed bolts adjusted to leave a clearance between it and the end of the distance-piece. Spring centring and a limited travel is thus afforded but there are no oil-pressure loaded plungers as in the smaller Ross gear and the Saginaw and no "power feel" is given.

The control valve is a separate assembly located above and parallel to the steering shaft and gets its movement by a lever with a spherical lower end engaging a groove turned in the cam and turning on a pin carried by lugs on the detachable valve body. A cylindrical upper end on the lever enters a socket spherically mounted in an eye, which is screwed into the end of the valve piston. Valve travel is twice that of the steering cam and this has permitted the employment of a fairly small valve, 1·3125 in diameter, on what is a large gear. It is designed for a controlled flow rating of 2·1 gal/min, giving lock-to-lock in six seconds.

Power piston area is 11·8 in^2 and the power lever arm is 3 in long, giving a hydraulic drop-arm torque of 25,400 lb-in at 750 lb/in^2 or 33,900 lb-in at 1,000 lb/in^2 pressure. These, obviously, are heavy commercial ratings and can be compared with a drop arm torque of about 9,500 lb-in at 750 lb/in^2 for the Saginaw gear on the Buick car.

Mention of power torque figures leads to consideration of the effect on the steering connections of the fitting of a power mechanism. In the hands of an inconsiderate driver there is no doubt that the average number of heavy loadings in a year's operation will be much increased. Expressed in another way, the driver will not trouble to ease his connections of load by keeping the vehicle in slow motion when making parking locks. Since most power steerings are designed to exert full parking torque for a very slight steering wheel rim force, it is also clear that, if the lock stops are on the steering swivel and not in the box a muscular driver can increase the *maximum* load by from 60 to 70 per cent by pulling hard on the wheel. On both counts it would seem desirable to review the stresses of every part of the steering linkage when changing over to power operation.

For those interested in power steering equipment that can be fitted to an existing chassis and use the standard manual steering box, both Ross and Vickers offer systems, primarily intended for commercial vehicles, in which the power mechanism comes in the steering linkage. A description of the Ross linkage-type gear may be of technical interest.

In this system the usual spool valve, provided with reaction rings acting under pump pressure to give centring and "power feel," is incorporated with the drag link. Flexible hoses from the valve assembly connect to the two ends of a separate cylinder, suitably coupled to the steering linkage, while two more hoses make the flow and return connections to the pump.

Such a system could be fitted in a few hours to an existing vehicle.

BRAKE ASSEMBLY

Developments at the Works of Automotive Products Co. Ltd.

MANY novel and extremely efficient developments have been incorporated in the various assembly sections laid out by the Brakes Division of Automotive Products Co. Ltd., Leamington, for the assembly of brakes for Austin A30 cars. In all there are three distinct sections; one for the assembly and testing of the brakes; one for the assembly and testing of the master cylinder; and one for the assembly and testing of the frame cylinder used in connection with the rear brakes.

In planning the various layouts, three main factors have been considered. The first was to reduce handling to a minimum, both in the delivery of materials to the assembly sections and in passage through the section; the second was to use the least possible amount of floor space for the specified maximum output; and the third was to provide sufficient flexibility to allow variations in output to be handled economically. It is almost needless to say that the overriding consideration was, of course, the quality of the final product.

Only one of the sections, that for brake assembly, could be in any way considered as suitable for a power driven conveyor, but examination of the problem quickly showed that the use of such a conveyor would be uneconomic even for full output and would also reduce the flexibility of the section for dealing with variations in output. It was therefore decided to plan a layout that would allow the parts to be passed manually from one station to the next, either by direct transfer where the stations were adjacent, or by chute where direct transfer was not feasible.

A noteworthy feature of all three sections is that work benches have been completely eliminated. In their place there are steel racks, fabricated from angle sections and designed to hold standard tote pans of components in positions convenient for the operators at the various work stations. All the component parts for the master cylinder and frame cylinder are delivered to the assembly sections in these tote pans; for the brake assembly all parts other than the back plates and the shoes are delivered in them.

This use of standard tote pans greatly reduces the amount of handling that is necessary. For example, parts that are produced in one of the Company's machining sections are loaded into tote pans in standard quantities at the end of the machining operation, and they eventually arrive at the assembly section in the same pans. Bought-out parts are loaded into tote pans in the receiving stores and are then passed to the assembly sections.

Brake assembly

The brake assembly section may be considered as comprising the machines for riveting the linings to the shoes, the machines for grinding the shoes, and the actual assembly stations. To minimize the amount of handling that is necessary, the riveting machines are located adjacent to the grinding machines so that the work can be passed down a chute from one machine to the next. Similarly, the grinding machines are located to allow the finished shoes to be passed down a chute to a position adjacent to the station at which the shoes are assembled to the back plate. The back plates are delivered in stillages to the appropriate station. All other component parts are delivered in standard tote pans. At most work stations the fixtures are mounted on steel plates welded to the steelracks.

On the Austin A30 the front brakes, see Fig. 1, are hydraulically actuated, and incorporate wheel cylinders, while the rear brakes are mechanically actuated. There is therefore need for slightly different assembly sequences, certain stations being used only for front and others only for rear brakes. Provision has also had to be made for dealing with different handed assemblies for the production of off and near side brakes. In general, the work fixtures are designed to accommodate both right and left hand assemblies, except for the hydraulic test on wheel cylinders for the front brakes. For this test it has been found advisable to have separate testing stations.

There are also two stations which do not lend themselves to inclusion in the actual assembly line. At one of these stations there is a foot press that is used for riveting the clicker spring to the expander body for the rear brake. This press is adjacent to the assembly line. Two fabricated angle steel structures are attached to it, one at each side, taking the component parts at one side and the sub-assemblies at the other. The second fixture not actually in the line is used for rolling the pistons for wheel cylinders to ensure that they are completely free from burrs.

To economize in floor space, double sided racks are used in the brake assembly section with working stations at each side. Passage of completed sub-assemblies from one side of the line to the other is effected by means of chutes which deliver direct to the appropriate station. On what might be termed the final assembly side, there is need to transfer certain assemblies to fairly widely separated stations. In one case this is effected down a chute running beneath the racks that carry the tote pans, and in another down a length of roller conveyor.

It is not intended to describe the complete assembly in detail, but rather only to draw attention to the manner in which the whole section is laid out to allow the work to be carried out quickly and efficiently. For example, at almost every station the work

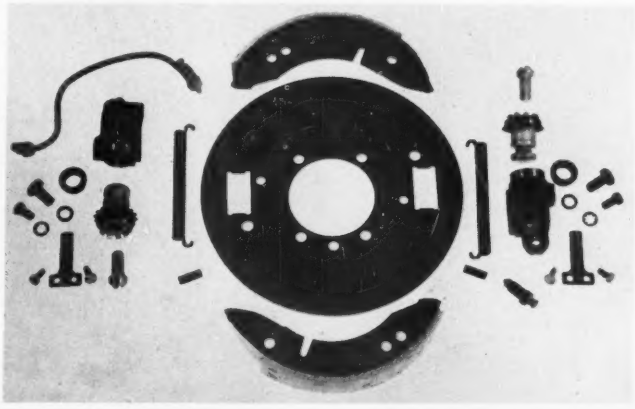


Fig. 1. Components for front brake assembly

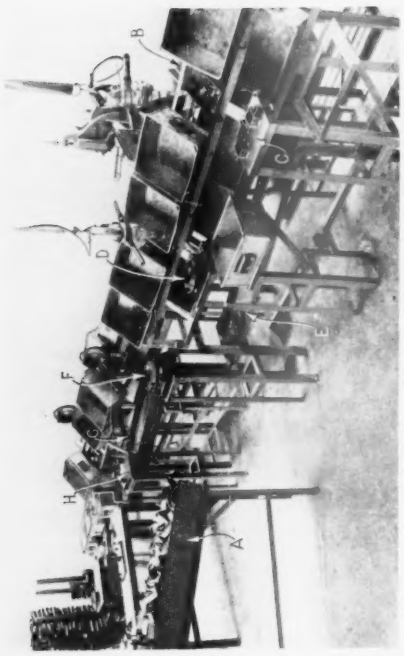


Fig. 3. Final assembly side of brake assembly section

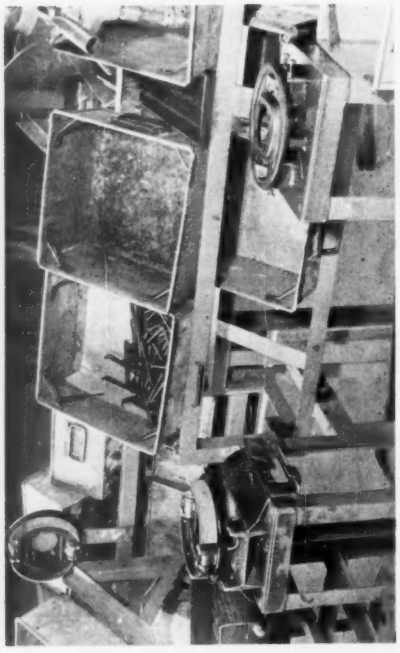


Fig. 5. Back plate assembly stations for front brakes

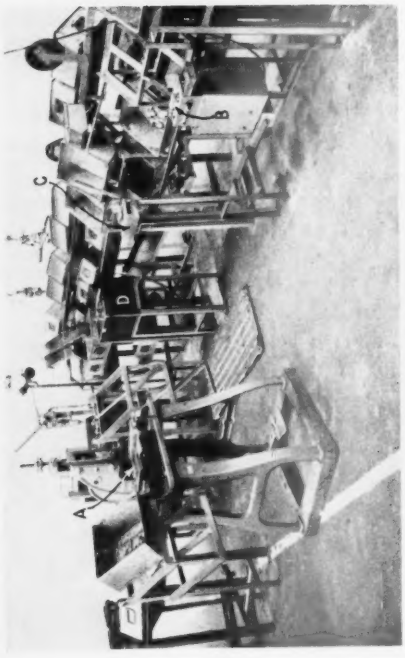


Fig. 2. Sub-assembly side of brake assembly section

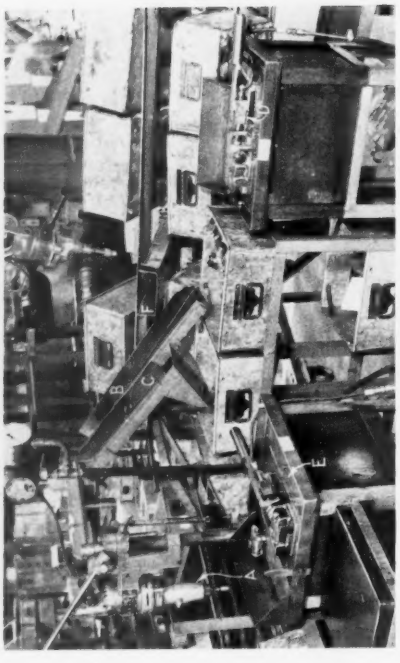


Fig. 4. Hydraulic test rigs for wheel cylinders

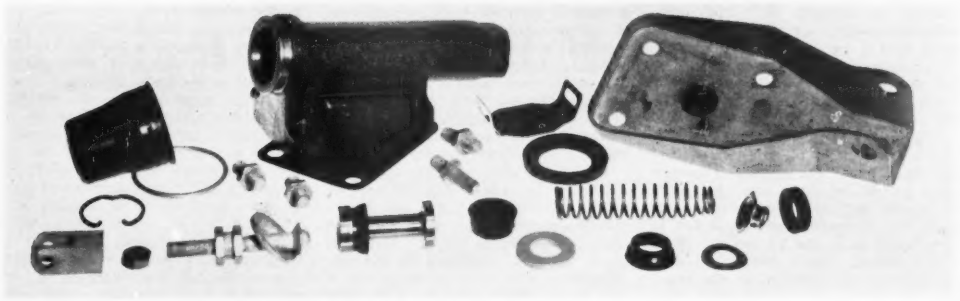


Fig. 6. Components for master cylinder

is placed in the fixture and held sufficiently securely without any tightening. Where screws or nuts are to be screwed in or tightened, the general practice is to use Wolf electric nut runners, suspended and counterbalanced in positions convenient for the operators. At one station, however, a special inverted nut runner is employed. It has two heads driven from a common source. Each head is recessed to take a screw, and it is only necessary to press the part into which the screws go for both to be driven home simultaneously.

Every wheel cylinder is tested hydraulically after the sub-assembly is completed. At one time this test was a comparatively lengthy operation, since it entailed making and unfastening screw connections. Now the wheel cylinder is merely fitted in a jig and all the necessary blanking is effected without a single screwed connection. The jig, and with it the cylinder, is then turned through 90 deg for immersion in a tank of hydraulic fluid. This test is now carried out in a time that conforms with the time cycle for other operations.

The general layout of the brake assembly station is illustrated in Figs. 2 and 3. Referring to Fig. 2, A is the foot press for riveting the clicker spring to the expander body, B is the two-spindle nut runner station for the clicker wheel and adjuster assembly, C the expander body assembly station, and D one of the hydraulic test rigs for the wheel cylinders.

Referring to Fig. 3, A is a chute from the grinding machine for delivering shoes and linings ready for assembly to the back plates; B is a chute from the hydraulic test station for delivering wheel cylinders to the front brake assembly stations C and D at which the wheel cylinders are assembled to the back plates; E is the chute for delivering back plate and wheel cylinder sub-assemblies to assembly station F where the lined shoes are assembled to the sub-assembly; and G is a roller gravity conveyor for carrying the front brake assemblies to the station where the bridge pipes, bleeder screws and blanking caps are fitted. All assemblies of components to the back plates for the rear brakes are

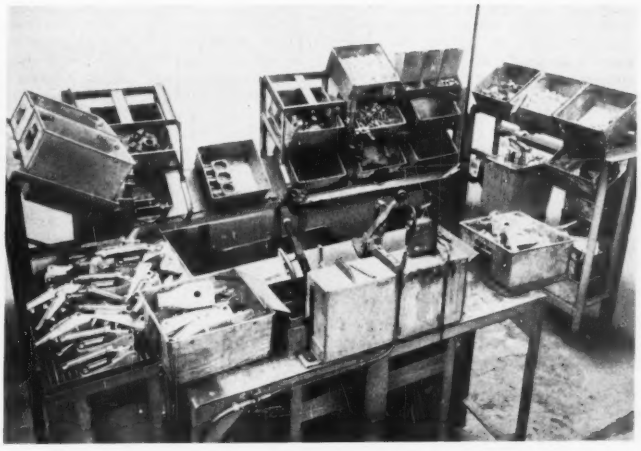


Fig. 7. Master cylinder assembly section

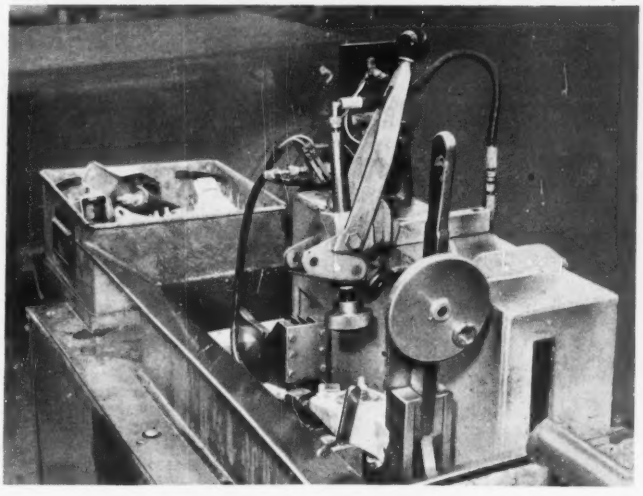


Fig. 8. Master cylinder hydraulic test rig

effected at station H. The sub-assemblies, which are produced on the other side of the line, are passed down a two-branch chute, one branch for left and the other for right-hand parts. Each branch delivers direct to station H.

Fig. 4 shows the two hydraulic test rigs for wheel cylinders. In this illustration, the clicker spring assembly with its electric nut runner is shown at A; the delivery chutes from station A to the hydraulic test rigs B and C are shown at D and E; and the chute for delivering tested wheel cylinders to the assembly station on the other side of the line are shown at F. The simple types of fixtures and the manner in which they are built into the structure are illustrated in Fig. 5, which shows at the right the station for the assembly of the wheel cylinder to the back plate, and at the left, the station for the assembly of the shoes to the sub-assembly from the previous station.

This assembly layout is a very marked departure from conventional practice, but it has already proved its worth. The floor space occupied is much less than would have been necessary under any of the earlier assembly arrangements, and we should estimate that the initial outlay has been less. Furthermore, should occasion arise, the angle iron structures can be readily adapted to a revised layout.

From the production engineer's point of view, probably the most attractive feature of this layout is the ease with which wide variations in output can be handled economically. Naturally, the most economic costs are obtained when the section works at maximum capacity with an operator at every station. When, however, through shortage of materials from outside suppliers, or for other reasons, the section must work at less than full capacity, the number of operators can be proportionately reduced (each operator working two or more stations) and the loss in efficiency is only slight.

This assembly system has also brought other advantages. One of the most important is that it allows specified outputs to be maintained with quite small buffer stocks, with the result that the investment in work-in-progress is kept low. As the policy of the Austin Motor Co. Ltd. is to carry on small stocks in relation to output, Automotive Products Co. Ltd. must be prepared to make frequent deliveries of relatively small, and perhaps fluctuating quantities. This condition could, of course, be met by building up stocks of finished brake assemblies, but this has not been necessary because the planned methods allow very quick response to any change in demand.

Compact assembly sections have also been devised for the master cylinder and the frame cylinder. A set of components for a master cylinder assembly is shown in Fig. 6, and the assembly section is illustrated in Fig. 7. Once again fabricated angle section racks are used for carrying standard tote pans of components. For the master cylinder there are only three assembly operations and one test operation. Of the assembly operations, it is necessary to mention only the first. At this station, which may be seen in Fig. 7, the baffle, gasket and cover plate are fitted to the cylinder. A special inverted four-spindle nut runner is used. The screws are placed in the runner heads, the cover and the gasket are placed over them, and the body is placed in position on top. A vee-shaped clamp is then pulled down on to the body, the nut runners and the screws start to revolve, and a light pressure is maintained on the body to drive it on to the screws.

The testing rig, which is shown in Fig. 8, is noteworthy. It completely eliminates the need for screwed connections and allows a complex test sequence to be carried out in a remarkably short time. The body is dropped into a base plate and a toggle link mechanism is actuated to seal off the back end. Another toggle link mechanism in which a self-opening valve is incorporated, is then applied to the top. Through this valve a pressure of 20-30 lb/in^2 is developed inside the cylinder to test the casting for porosity. After the porosity test, the top valve is removed and the fluid tank is raised to fill the cylinder with brake fluid. Air is then passed through the end fitting to ensure that the vent hole is clear. By means of a calibrated hand-wheel, the piston is then moved forward 0·030 in; the air must not be cut off until this degree of movement has been reached. At this stage, the back toggle mechanism is disengaged and the piston is operated by hand lever. After several strokes, the cylinder is again filled with brake fluid and the back toggle mechanism is again engaged. The air pipe line connected to the back toggle mechanism is blanked off, and the piston is again operated until a stop light switch connected with the toggle mechanism causes a small electric bulb to light up. This gives proof of correct sealing and completes the test.

MIRA REPORTS

Recent Releases for General Circulation

SEVERAL reports on different subjects that have been investigated by the Motor Industry Research Association have now been released for general circulation. Among them are included:

Report No. 1951/2 on the *Fatigue Strength of Cast Crankshafts*, which deals with the influence of some special design features on bending fatigue strength. The investigations were concerned with the following design features: web depressions, journal bores with well-rounded ends, journal bores with ends reduced in diameter, chamfered webs, eccentric journal bores and slots machined through the web.

Some of the designs were subjected to bending fatigue tests and others were more conveniently examined by stress measurement. None of the designs gave increases or decreases of strength of appreciable magnitude. The best improvement indicated was for the journal bores with reduced ends (barrel-shaped bores). It was in the order of 10 per cent, which is negligible compared with the improvements in strength obtainable by increasing the fillet radii. Some of the designs did, however, have appreciable effect on crankshaft bending flexibility. For example, web depressions increased the flexibility by about 50 per cent.

Report 195037 on the *Bending Fatigue Strength of Gear Teeth*. This report is a comparison of some typical gear steels. The steels were 1½ per cent Ni-Cr-Mo oil-hardening, both plain and leaded; 4½ per cent Ni-Cr air-hardening; 3 per cent Ni-Cr case-hardening; 4½ per cent Ni-Cr case-hardening, both plain and leaded; and 3½ per cent Cr-Mo-V nitriding.

Gears in the various materials were compared on the basis of their mean "S-N" curves, where S is the nominal bending stress and N the number of stress cycles to tooth fracture. The case-hardened steels proved superior in fatigue strength to the through-hardened steels, nitrided steel gears were intermediate between these groups.

The introduction of lead to improve machinability reduces the fatigue strength of both through-hardened and case-hardened steels. In through-hardened steels lead has an adverse effect both in the region of finite life and on fatigue limit. In case-hardened steels, the effect becomes more pronounced as the number of stress cycles increases. Indeed the leaded case-hardened steel showed no fatigue limit up to 10 million cycles.

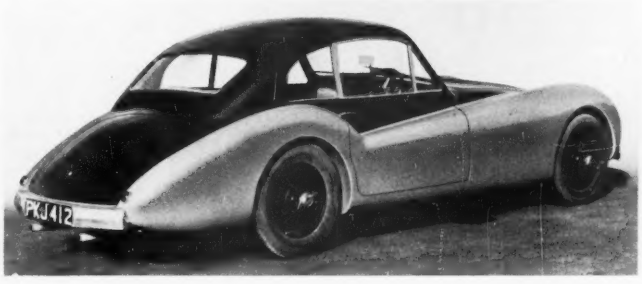
Price indices of the various materials are included in the report, but have not been directly related to fatigue limit, since in comparing materials for gear application it may be necessary to consider both the fatigue limits and strengths at a finite number of cycles.

Copies of these reports, price 10/-, may be obtained from The Motor Industry Research Association, Great West Road, Brentford, Middlesex.

A MAGNESIUM BODY

An Experiment With a Lightweight Material

ESSEX AERO Ltd., of Gravesend, recently demonstrated the prototype of a magnesium bodied sports coupé. The object in designing and making this car was to show that it is practicable to form magnesium sheet into the complex shapes required for the modern car. From the accompanying illustration it can be seen that their objective has been attained. However, several relevant questions still remain unanswered. Magnesium alloy sheet is somewhat more expensive than aluminium, and it seems likely that the methods of fabrication employed would also be costly. The properties of this material, such as strength, fatigue resistance and corrosion resistance are not so good as those of aluminium. Nevertheless, so far as these properties are concerned, little trouble is likely to be experienced in a car designed and manufactured on lines similar to the demonstration vehicle.



The Essex Aero magnesium-bodied car

Because of its light weight, magnesium is an attractive material from which to manufacture sports cars. It is approximately two-thirds of the weight of aluminium. Moreover, it is, at the moment, in rather more plentiful supply than aluminium. A question that is often asked about this metal is how easily will it ignite during processing. Very small pieces, such as magnesium ribbon or filings will burn readily. In sheet or cast form, however, it must be kept at a temperature of at least 600 deg C if combustion is to be maintained. The freedom from fire danger is clearly shown by the fact that the forming process consists in alternately heating the metal by blow-lamp to a temperature of about 300 deg C and panel beating it.

Corrosion is a serious problem with this material, particularly with copper to magnesium, or brass to magnesium joints, but chromium or cadmium plating is an effective corrosion inhibitor. Alternatively, both sheet and cast magnesium may be given a chromic acid treatment which, in conjunction with varnish, effectively prevents corrosion. So far as this particular vehicle is concerned, the treatment given was as follows. After fabrication, the body was dipped in a dilute nitric acid bath, and then in boiling water. This dipping process was carried out in order to remove all traces of welding flux, grease, etc. Next, it was given a chromic acid treatment followed by a coat of Varsyn varnish and it was then stoved. Finally, after applying Bergers grey primer, the cellulose finishing coats were sprayed on.

Argonarc welding was employed for fabrication. This process gives rise to little distortion, as the heating effect is very local. Moreover, the argon shroud prevents the formation of oxide, and so enables a sound joint to be made, while calling for the application of less skill than most of the other welding processes. It is claimed in the U.S.A. that magnesium sheet may be spot welded. However, the clean joints that are necessary for such a process would probably give rise to corrosion troubles.

The demonstration vehicle was based on an Allard J2X chassis which, to accommodate a fully streamlined body, was extended at both ends. A light alloy structure at the rear formed the main extension, 2 ft 9 in long. Further modifications were made to the chassis to increase by 1 ft the length between the driving seat and the rear wheels. These modifications were effected by moving the pedals forward, and dropping the floor to about 2 in below the top of the chassis frame. A saving of weight was obtained by replacing the steel petrol tank with a tank made of magnesium; the relative weights of the two were respectively 39½ lb and 15½ lb.

It is, of course, easier to give a well streamlined shape to a coupé than to a four-seater saloon. Full advantage was taken of this fact in the demonstration car, which had only an occasional seat behind the two front seats. The back light is not unusually large as is the case when a fully streamlined design has been adopted, because a slightly stepped rear end style has been chosen. Bucket-type front seats were employed. They were made from 16 s.w.g. magnesium alloy sheet.

The body shell, with doors, weighs 140 lb, and is welded together in one piece. For the most part it is of 16 s.w.g. D.T.D.118A magnesium alloy sheet. The windscreen pillar, door sill and reinforcement around the door opening are of 12 s.w.g. sheet. Most of the hardware is made from magnesium alloy, either ¼ in or $\frac{5}{16}$ in thick. These items include the bonnet and door hinges, body mounting brackets, and boot lid hinges. The bumpers are fabricated by welding from ¼ in thick magnesium alloy sheet, containing 7 per cent aluminium. They are formed into a channel section which is very solid and serviceable, and yet they weigh only 8 lb each. The grille is made from 10 s.w.g. vertical strips welded to ¼ in thick transverse strips.

An interesting feature of the body is that it may be completely removed from the chassis in about four minutes, and only four men are needed to lift it clear. It is for this reason that the one-piece construction has been employed. No doubt this would appeal to some sports car enthusiasts since it leaves the chassis completely clear for adjustments and tuning. Moreover, when the body has been lifted clear, the front seats and the floor remain on the chassis, so that it may still be driven. Nevertheless, this unit body construction would appear to present difficulties, so far as repair of damage to wings, etc., is concerned. The removal sequence is as follows:—

1. Slacken front bumper bolts, and slide the bumper forward.
2. Disconnect electric wiring.
3. Disconnect one heater pipe.
4. Remove the body mounting bolts.
5. Lift the body vertically.

Six body mounting points are employed, the attachment brackets being welded one on each side of the transverse hoops of the chassis structure. Thus, there are two mountings at the front, two at the centre, adjacent to the door pillar, and two at the rear, over the back axle. The body shell, which is suitably reinforced at these points by a welded internal structure that carries the body brackets, is

bolted to the frame by $\frac{1}{2}$ in diameter bolts used in conjunction with Silentbloc resilient mountings. The top bearings of the door hinges are incorporated in the pair of body mounting brackets adjacent to the front pillars, while the lower hinges are entirely separate.

A considerable amount of Argonarc welding must have been employed in the construction of the body. Not only were the separate portions of the panelling butt-welded together, but also the stiffening sections round the edges of the boot and bonnet lids, and round the openings, were made of magnesium channel section, with its upper flange butt-welded to the panel. Butt-welds are employed in preference to lap welded joints because they eliminate the possibility of the onset of corrosion due to the collection of moisture between the overlapping faces. With this material every possible precaution must be taken against corrosion, and scratches to paintwork would have to be carefully treated very soon after they occur.

There is little doubt that the methods of construction employed on this prototype are the best, so far as prevention of corrosion is concerned, but they are expensive. Other methods, such as the employment of dutch bends around door openings and boot and bonnet lids, etc., might be used, since magnesium becomes pliable when heated. Whether such methods of construction would be satisfactory in practice remains to be seen. It would appear that the only attraction that this material has to offer to the customer is an extremely light weight, but it is doubtful whether even the most enthusiastic sports car owner would be prepared to pay a price very much in excess of that for an aluminium body.

STEAM SPRAYING

An American Development

STEAM spraying for applying paint is an entirely new development from America. The process has been developed by E. I. Du Pont de Nemours and Co. It is effected by using super-heated steam in place of compressed air. Steam spraying offers many advantages to manufacturers. Some of them are outlined below.

1. It reduces costs by making more efficient use of the paint. When spraying is done by compressed air at 50 to 80 lb/in^2, spray efficiency rarely exceeds 65-70 per cent, and there are considerable losses. These losses occur because particles of paint are prevented from reaching the surface of the component. On the other hand, in steam spraying, the steam dissipates immediately on release, enabling the material to concentrate upon the work. Thus overspray is reduced and there is a saving of 10-20 per cent in the amount of material used.

2. It cuts finishing time, because with steam spraying the operator can apply a film thickness in one coat that is equal to two coats sprayed by the more conventional method.

3. Labour costs are reduced, because fewer coats have to be applied. Reduced overspray also reduces paint losses, and the spray booth does not need to be cleaned so frequently.

4. Through the action of the atomizing steam, the paint is applied at a lower viscosity at the point of atomization. It is therefore possible to use material of higher viscosity than is used in compressed air spraying. This allows the application of heavier coats without sagging or running.

5. Cost is reduced on thinners because higher viscosity materials are used.

6. Danger of sags, runs, etc., is greatly reduced, because after atomization the material rapidly cools back to room temperature. With an appropriate change in viscosity, it is possible to apply heavier coats than by using compressed air.

7. Since fewer coats are applied, production is increased.

8. It costs less than air spraying because generally steam can be produced more cheaply than compressed air.

9. It is less hazardous. As overspray is reduced and solvent fumes are less, working conditions are improved.

10. Rejects for pinholing, normally caused by entrapped air, are reduced to a minimum, and orange peel effect is not so pronounced. The result is a far better surface and a higher quality product.

11. Steam spraying can be fitted into any existing installation, providing that steam is available at a pressure of 125 lb/in^2. The danger is that condensation may come through the lines, beyond the capacity of the superheater to dry it.

The disadvantages are:—

1. To avoid excessive hose length and consequent cooling of the steam, which must be under constant pressure, the steam source must be located close to the operating stator.

2. Few plants have superheated steam supplies, but equipment is available to provide steam at the required pressure and dryness.

Steam spray guns

The gun has two outstanding features; it weighs no more than the ordinary spray gun; and there is practically no heat transfer between the head of the gun, where both steam and material inlets are located, and the body of the gun. Owing to excellent insulation and heat radiating design, and the use of a built-in finger guard, low trigger and handle temperatures are assured.

The hose support is positioned to give fine balance to the gun. As the gun is the bleeder type that allows steam to escape continuously during operation, condensation in the steam lines is eliminated. Special rubber hosing is needed to withstand the prolonged temperatures up to 400° F.

The pressure gauge which indicates the atomizing steam pressure is located between the outlet of the superheater and the entrance of the steam hose. A thermometer is also fitted between the superheater and the entrance of the steam hose. This thermometer indicates temperature of the superheated steam.

To eliminate water particles present in ordinary saturated steam, which might cause blistering or pinholing in the finish coat, it is necessary to superheat the steam to convert all water content to dry steam vapour. It is recommended by the Brake Shoe Company of America that a 3 kW superheater capable of serving up to two spray guns, is the minimum size unit suitable for most applications. These units are explosion proof and are normally used at 230 voltage A.C.

A pressure regulator duplicates the purpose of a regulator in an air line, but in this case it controls the pressure of the steam entering the superheater. It operates by diaphragm pressure against spray pressure. The regulator is located on the inlet side of the superheater, since by controlling the steam pressure at this point an accurate reading may be obtained from the pressure gauge on the outlet side. A separator is required to remove any water that may be present in the steam, and return it to the generator, or to the drain.

There is little or no danger from burns, although in the cause of safety it is recommended that operators wear gloves. The superheated steam cools so rapidly that the operator can pass his hand through the spray at a distance of 12 to 15 in from the gun without any discomfort.

PRESS PLANT

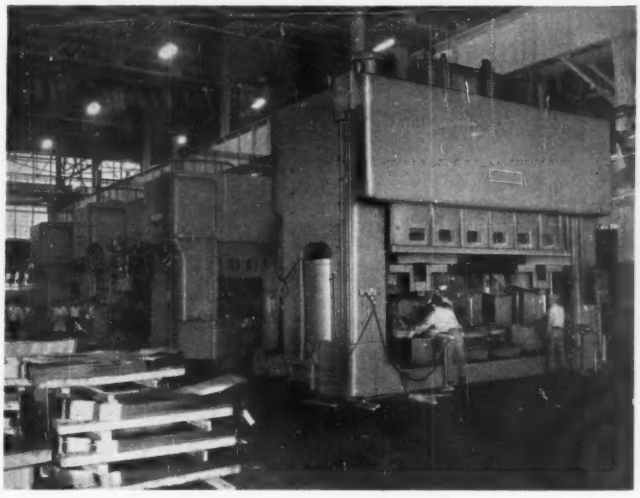
An Interesting Development by the Chrysler Corporation

A NEW and very interesting press plant has recently been completed and put into operation by the Chrysler Corporation. It is located at Nine Mile and Mound Road, Detroit, Michigan, U.S.A. There are two outstanding features in this new plant. The first is the provision made to allow the original installation of heavy presses to be carried out with comparative ease. The second is the ease with which heavy presses can be moved from one location to another to suit production requirements.

Before the development of this plant, the presses were designed and installed with most of their operational parts above floor level, and a pit, sometimes to a depth of 20 ft, had to be excavated under each press location to take the foundations. Once a press was set into position, servicing was difficult and any change of location was a lengthy and expensive operation.

In the new plant an open basement is provided underneath the areas in which the presses are located. The machines themselves can now be designed with most of their working parts below floor level so that there is not the same press height above the floor. In addition, the basement provides a clear space where workmen can get at the mechanism of each press quickly and easily for servicing and repairs.

In the conventional press plant the transfer of a large press from one location to another involved a repetition of the original procedure of excavating a hole and preparing a concrete foundation. To move a press, or to add a new one at the Nine Mile plant, it is only necessary to remove sections of the 4 in laminated oak flooring surrounding the presses on the working floor. Then, overhead cranes can be used to move the press in sections from the original to the new location.



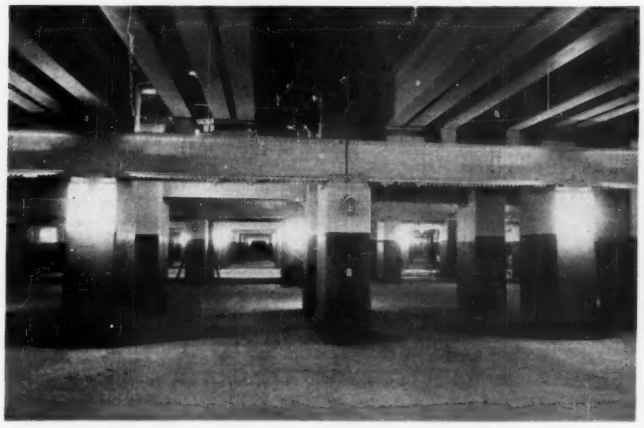
Heavy presses in use at the Chrysler Corporation Nine Mile Press Plant

Some indication of the problem involved in moving large presses may be obtained from a consideration of their dimensions and weights. The two largest presses now in use at this new plant weigh 440 tons each. In operation they accommodate dies weighing 75 tons and exert a pressure of 900 tons on the metal being formed. Each press occupies a floor area 12 ft × 24 ft and stands 15 ft above floor level and 17 ft below it.

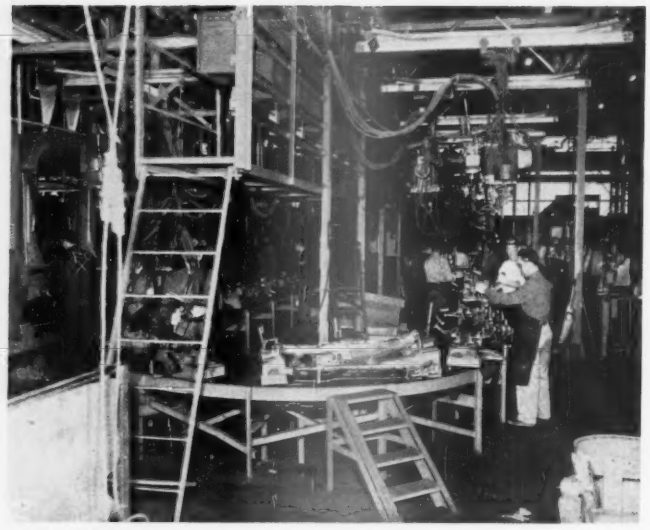
The factory building is 704 ft long × 462 ft wide. A raw material receiving area is located along the north side. It has a clear span of 100 ft and is served by two overhead bridge-type cranes of 50- and 20-tons capacity. The crane rails are 32 ft and the underside of the roof trusses 42 ft above the floor. Raw material is delivered both by rail and by truck. The press area and die shop, the assembly and welding area, and the storage and shipping area are along the south and west sides of the building.

The actual press area measures 240 ft × 640 ft. Beneath it there is a pit, or basement, 17 ft deep and with a superficial area of 176 ft × 480 ft. Owing to the soil conditions, a heavily reinforced mat, 3 ft thick, was installed under the whole of the basement. Concrete piers, at 40 ft centres one way and 13 ft 4 in the other, support the 176 ft continuous lower tier press girders. Pairs of these lower tier girders, spaced 30 ft apart, form 12 separate aisles of press supports. Between each pair of press supports there is a 10 ft wide gangway.

In each of the 12 press aisles, the ends of the upper tier press girders rest on the top of the lower tier



Part of the basement below the press area. The press mechanism can be serviced from the basement.



Spot welding centre pillars in the Nine Mile press plant

girder. The upper tier girder comprises two 36 in girder sections with 20 in top and bottom cover plates. Slotted holes are provided in the cover plates to allow the press anchor bolts and the presses to be moved to any desired lateral position. In addition, the upper tier girder can be moved to any desired position lengthwise in the press aisles. Floor framing and 4 in laminated oak flooring is provided between and around the presses. The press area is served by 50-ton cranes on eight 80 ft × 240 ft runways. The receiving area is at right angles to the press aisles. A 16 ft wide ramp and stairs connect the work floor with the basement.

An area south of the presses is used for assembly and welding operations and for finished storage. In the same area there are repair shops, tool stores and shop offices. The die shop is located along the west side of the building. Despatch of the finished product is through a truck dock, 120 ft wide, located at the east end of the finished storage area. Two large balers are installed for compressing and baling scrap and trimmings. The bales are dropped on to a continuous belt conveyor below the floor. On this conveyor they are carried outside the building to be discharged automatically into open railway trucks.

Because of the extremes of temperature encountered in Detroit, heating and ventilation are very important. Steam for heating the building and for process requirements is supplied at 150 lb/in^2 pressure from the present boiler house of the Dodge truck plant. These steam and return services run through the present truck plant and then, through a long underground tunnel, to the press plant.

Thermostatically - controlled, overhead down-blow units heat the press shop. In addition, each large door has a blower type heater that automatically turns on when the door is opened. This supplies a curtain of warm air across the full width of the door. These heaters are all supplied with steam at 30 lb/in^2. In addition to the unit heaters there are four winter supply fresh air units which supply 80,000 c.f.m. of heated outside air to the buildings.

In summer, the ventilation of the press shop is effected by means of unit type ventilators, both supply and exhaust, on the roof. They provide approximately ten air changes per hour over the entire area, to keep the air clean and fresh to give good working conditions.

Great care has been exercised to make maintenance of the presses as easy as possible. All electrical connections between each press and its associated control devices are made from carefully engineered layouts. Conduit and wiring connections between the devices have been planned to follow the most direct and logical route. All similar presses have identical conduit and wire inter-connections with all device terminals tagged to conform to the manufacturer's control diagrams.

All the large presses are equipped with four multi-point receptacles located at each corner, and designed to function with one standard type of portable control stand. One, two, three or four operators can work on a single press in complete safety if a standard control stand is plugged in for each operator. One "stop" and two "run" buttons are provided on each control stand, and the press will not start until all "run" buttons are depressed. A special shorting plug is also provided at each control station to permit control of the press from one position.

Automatic scrap baling machines

Metalastik Heavy Duty Shackles

A PARAGRAPH in the review of the Commercial Show, suspension section, published in the November, 1952, issue of this journal, may have created an impression that vehicle makers who had hitherto fitted Metalastik heavy duty shackles have now reverted to grease-lubricated spring joints. The facts are, of course, otherwise. Not only have Leyland Motors Ltd. retained these shackle pins in all the models on which they were originally introduced, but they are now also incorporated in the B.U.T. trolley-bus chassis, and certain Guy, Bristol and Continental chassis.

FATIGUE OF METALS*

Orowan's Theory and Its Practical Implications

THE nature of the spread of a fatigue failure has been explained by Gough[1] in the following terms :—

At some point, before the yield point can be determined by external measurement, local yield actually takes place in a crystal or at a grain boundary. This is because there are many imperfections, even in a pure metal, which give rise to stress concentrations. Owing to the applied oscillating stress, considerable slip takes place at the edges of the crystal, and the quantity of slip increases until the edges of the grains under the microscope appear to be covered with black patches, which are really a large number of highly concentrated slip bands. If these black areas spread more than a certain amount, then the fatigue crack is propagated and fracture will result, but if the applied stress is not too great, the slip bands will eventually cease to spread; the metal stabilizes itself, and failure will not occur.

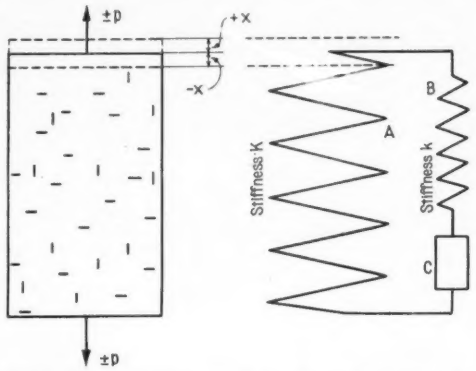
Orowan's theory

A theory of fatigue failure has been put forward by Orowan[2] which, in the author's opinion, is the most compatible with the experimental evidence available. An understanding of the conceptions of this theory is most helpful in estimating the significance of certain fatigue phenonema.

A metal contains a large number of inherent imperfections. In Fig 1 these have been represented as being equivalent to a series of small cracks. Under tensile loads they will cause stress concentrations at the ends of the cracks, and a certain amount of plastic deformation will take place.

The metal may be regarded as equivalent to the spring system shown in Fig 2. In this diagram, A is a very strong spring, of stiffness K, representing the main body of the metal, and B is a weak spring in series with a plastic body C. B and C are parallel with A, and represent the stress-strain conditions at the imperfections. They contribute very little to the load carrying capacity of the system, and whatever changes take place in C, the deflection due to load $\pm P$ will be $\pm X$ where $X = \frac{\pm P}{K}$. It is the alternating strain of $\pm X$ on sections B and C of the system which has to be studied.

Let the stiffness of the spring B equal k, and the plastic strength of C be f. Then the spring B will deform $\frac{f}{k}$ before C suffers any deformation. If $\frac{f}{k} \geq X$, C will not deform at all, but if $\frac{f}{k} < X$, C will suffer plastic deformation at every application of the load. However, should C work harden as deformation continues, the value of f will increase and eventually a value will be reached when $\frac{f}{k} = X$. Then no further deformation will occur under the alternating strain $\pm X$. If, however, C cannot work harden to this degree, the deformation eventually will be so great that C will fracture. As B and C together represent the highly stressed area at the imperfections, the fracture of C is analogous to the propagation of a crack from the ends of one or more of these imperfections, that is, the initiation of a fatigue failure.



Figs. 1 and 2. The main body of the test specimen, Fig. 1, is represented by the spring A, Fig. 2, and the imperfections by B and C

The whole process may be represented graphically as shown in Fig. 3. In this diagram the curve OD represents the work hardening characteristics of the metal, that is, the stress strain diagram to some scale, and OE is the loading line for the spring B, the slope of which, of course, equals k. Mark off along the abscissa a distance to represent X, and at that distance from the origin draw the vertical line FG cutting OE at H. Finally, through D draw a horizontal line cutting FG at J. If H lies below J, then it means that the metal around the imperfections can

* *From a paper presented before the Institution of Production Engineers by Professor J. A. Pope, Ph.D., D.Sc., Wh.Sch., A.M.I.M.E.*

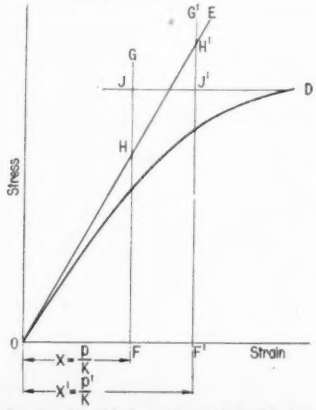


Fig. 3. Graphical representation showing the liability of a specimen to crack propagation

NOMENCLATURE

$\pm P$ = Range of oscillating direct stress.
$\pm P_f$ = Range of oscillating direct stress at fatigue limit.
P_m = Mean value of direct stress.
P_r = Fatigue limit when $P_m = 0$.
P = Oscillating tensile load.
P_m = Mean tensile load.
P_y = Yield strength (tension).
P_u = Ultimate strength (tension).
q = Shear stress.
q_y = Yield strength torsion.
q_u = Modulus of rupture for torsion test.

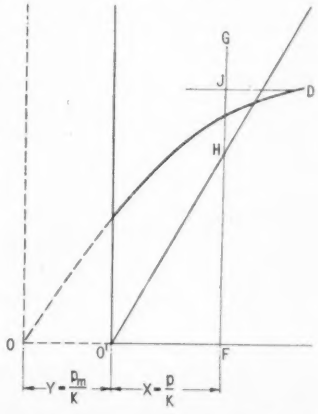
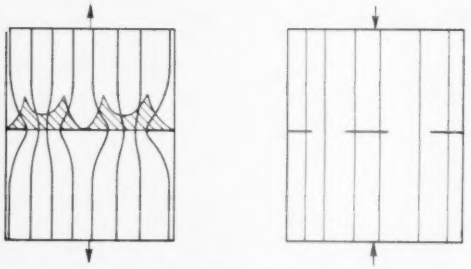


Fig. 4. The effect on the diagram, Fig. 3, of the application of a mean stress other than zero



Figs. 5a and 5b. The effect of stress raisers. (a) under tension, (b) under compression

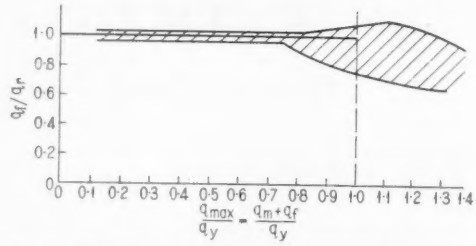


Fig. 6. Effect of mean stress on fatigue strength of polished specimens. The shaded portion is the area covered by experimental points

eventually work harden to such an extent that no further deformation will take place and the metal becomes stabilized. This is the case for J and H. If, however, the applied load on the specimen is increased from P to P^1, the deflection increases from X to X^1, where $X^1 = \frac{P^1}{K}$, and the points F^1, J^1, H^1 and G^1 may be obtained, Fig 3, and in this case H^1 lies above J^1, and fracture would occur. Thus the load $\pm P$ would not produce fatigue failure, but the load $\pm P^1$ would exceed the fatigue limit and failure would result. Fig 3 is, of course, qualitative rather than quantitative, but it gives a useful picture of the fatigue phenomenon.

Applying a mean stress

It is interesting to establish the theoretical effect of a mean stress according to Orowan's theory. The effect of applying a mean stress is simply to move the axis of the curve over a distance Y, Fig 4, where $Y = \frac{P_m}{K}$, P_m being the mean load. It follows therefore, from Fig 4, that the range of stress that may be safely applied to a metal should be independent of the mean stress, provided $P_m \pm P$ does not exceed the static fracture strength of the metal, since, although the axis of the curve is moved to the right by the application of a mean stress, the relative positions of points J and H remain unchanged. Actually, this is true for polished specimens subjected to torsional stress.

Another factor dominates the picture when compressive mean stress, or an oscillating compressive stress is considered. In many ways, the imperfections in a metal have an effect equivalent to minute cracks. Under tension, a crack is a stress raiser, since the stress cannot "flow" through the crack, but must "flow" round it and cause stress concentrations at the ends of the crack, Fig 5a. Under compression, however, very fine cracks are able to transmit stress, and therefore do not act as stress raisers. This being the case, the fatigue strength in compression should be much greater than in tension; this, in fact, is the case. Indeed, it is the fundamental reason for the improvement of the fatigue properties of metals by shot peening, surface rolling, nitriding. It is also the reason for the superior fatigue qualities of rolled screw threads as compared with machine cut threads.

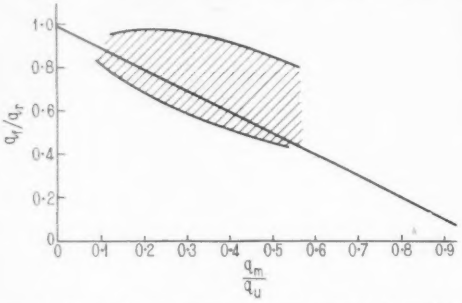


Fig. 7. Effect of mean stress on fatigue strength of notched specimens

Referring again to Fig 3, it will be noted that the strength at point D, that is, the rupture strength of the metal, is one of the main factors in fixing the fatigue strength. It follows then, that if a relationship exists between the fatigue strength and any one of the static properties of the metal, that property should be the true static rupture strength. Efforts to correlate the fatigue limit with the yield strength or the ultimate strength have been made without success. This is not surprising since Orowan's theory indicates that the fatigue limit is not related to the yield strength and, fundamentally, the ultimate strength is almost a meaningless quantity, related to the rate of work hardening rather than to anything else.

Unfortunately, although the results of a very great number of fatigue tests are available, the true static rupture strength of the metal is seldom given, and a huge fatigue testing programme would be necessary before the correlation would be possible between fatigue strength and the true static rupture strength.

The effect of mean stress

Smith[3] has carried out an extensive

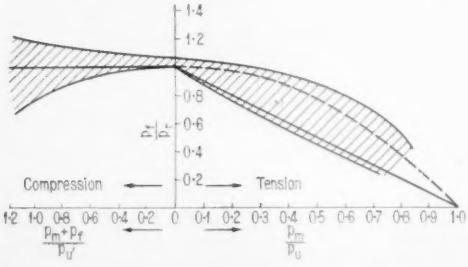


Fig. 8. Effect of mean stress on tensile fatigue of polished specimens—Smith

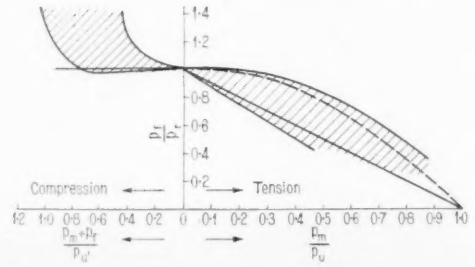


Fig. 9. Effect of mean stress on tensile fatigue of notched specimens—Smith

survey of all the important fatigue results, and has endeavoured to derive empirical relationships between fatigue strength and mean stress for various systems of stressing, and for both polished and notched specimens. Figs. 6, 7, 8 and 9 are derived from his results. There is a considerable amount of scatter in the fatigue results, and the shaded areas on these diagrams represent the areas covered by the experimental points. The parameters plotted vary according to the stress system under consideration, and the reader is advised to note the nomenclature carefully when studying these diagrams. The results for notched specimens are most important since a notch effect is present on anything that is not polished. Furthermore, most manufactured parts have unavoidable discontinuities, e.g. oil holes, and grooves. In practice the polished specimen condition is the exception rather than the rule.

Fig 6 shows the effect of mean stress upon polished specimens subjected to torsional fatigue. It will be noted that the range of alternating stress which may be applied is independent of the mean stress until the yield stress has been reached, that is, $\frac{q_{max}}{q_y}=1$. This is in accordance with the predictions based on Orowan's theory. The same does not apply to notched specimens, Fig 7. With these the safe range of stress decreases as the mean stress increases. For both polished and notched tensile specimens the safe range decreases as the tensile mean stress increases, Figs 8 and 9 respectively. When the mean stress is compressive, the safe range of stress remains constant for polished specimens, while for notched specimens the value of the safe range of stress increases rapidly as the compressive mean stress is increased. This is in keeping with the conclusions drawn from theoretical considerations. Some experimenters have found, even when a polished specimen is used, that there is a marked improvement in fatigue strength when the mean stress is compressive. Fig 10 shows the results obtained by Brown[4], for normalized mild steel. It will be noted that the fatigue limit was equal to the ultimate when the mean stress was compressive. This high value cannot be used in practice because deformation will occur and, since most engineering parts must be designed for freedom from deformation, in this case the yield strength becomes the limiting factor. The effect of a compressive stress is extremely important in connection with the surface treatment of metals.

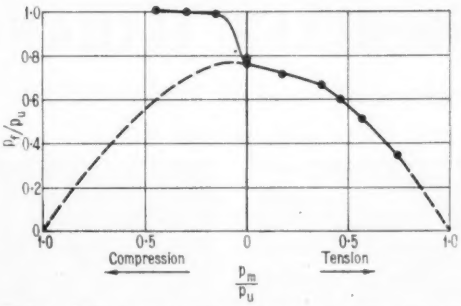


Fig. 10. Effect of mean stress on the tensile fatigue limit for normalized mild steel

Fig 11. Effect of stress concentrations on torsional fatigue strength

Types of fracture obtained in torsional fatigue

The actual shape of the fatigue curve obtained, i.e. stress plotted against reversals to fracture, will depend upon the stress system, that is, the value of the maximum principal stress p relative to the maximum shear stress q. In Fig 11, curve AB is for the condition $p=q$, and curve CD for $p \geq 2q$, that is, AB is for pure torsion without stress concentrations and CD either for tension, with or without stress concentrations, or torsion with stress concentrations.

It will be remembered that in pure torsion the maximum tensile stress, "p", is equal to the maximum shear stress "q", and that the latter acts on planes parallel, or at right angles, to the axis of the specimen. A fatigue crack will develop on one of these planes if the stress is above the fatigue limit, and there are not initial stress concentrations other than those intrinsic to the metal, but this crack will at once cause a stress concentration, which in turn will create a tensile stress $p > 2q$, i.e. more than double in value. Thus, the stress that cuts its corresponding fatigue curve at least reversals, will determine whether the fracture continues to follow the maximum shear plane or the helicoidal plane of maximum principal stress.

For example, consider stress q_1, point L, Fig 11. When the crack occurs the tensile stress will equal $2q_1$ which is well above the tensile curve CD, point M, with the result that the fracture will follow the helicoidal planes of maximum tensile stress, Figs 12a and 12b. If, however, a stress q_2 is applied point N, Fig 11, then, consequent upon the formation of the initial crack, the principal stress p is doubled, becoming equal to $2q_2$, point 0. In this case 0 lies below curve CD and the fracture will be circumferential as shown in Figs 12c and 12d.

This change in mode of fracture has been obtained by the author when testing hardened and tempered carbon steels. If the fracture is of the type shown in Figs 12a or 12b then the metal has been stressed well in excess of its fatigue limit; if it is of the type shown in Figs 12c or 12d the stresses have been lower. When the fatigue crack is initiated by the principal stress, the type of fracture will be that shown in Fig 12e.

The effects of surface conditions and treatment upon fatigue strength are also described. These are most important in bending and torsion.

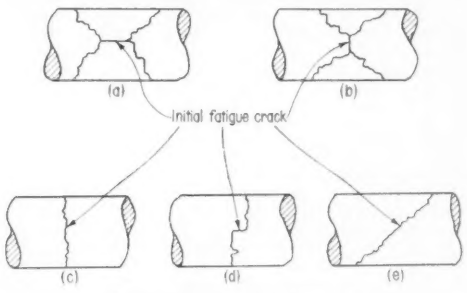


Fig. 12. Various types of fracture for pure torsion

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1. Gough, H. J. "The Fatigue of Metals".
2. Orowan. Proc. Roy. Soc., A171, 1939.
3. Smith, J. O. University of Illinois Bulletins No 334, and No 316.
4. Brown, R. M. Inst. of Engrs. and Shipbuilders in Scotland, Vol. LXXI, 1927-28.

CRANKSHAFT BALANCING

Methods Used by Daimler-Benz

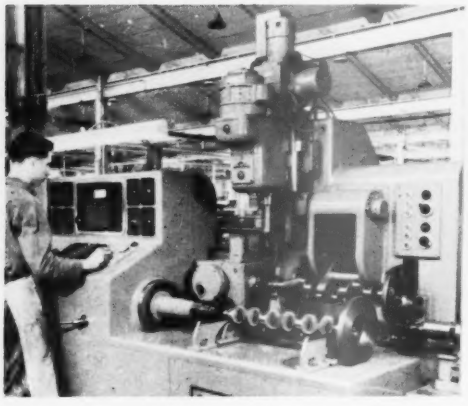


Fig. 1. Schenk crankshaft balancing machine

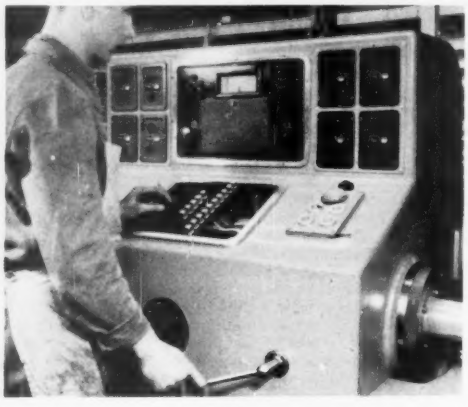


Fig. 2. The control unit

A MOST interesting balancing machine is now employed for balancing the crankshafts of the new Daimler-Benz engines. It is so designed that the complete balancing operation is carried out by one man. Furthermore, while one shaft is being rotated to determine the amount and position of the weight to be removed to give dynamic balance, the shaft tested immediately before is being drilled to give true balance. The equipment is shown in Fig. 1.

A crankshaft is delivered to the balancing machine on an electric overhead trolley and is dropped into the front bearings. It is then coupled to the drive and rotated at 3,000 r.p.m. In a matter of seconds the machine automatically determines the out-of-balance at two points on the crankshaft to within one gramme for weight and 2 deg for angle. This information is reported electro-dynamically to the control apparatus, shown in Fig. 2.

As soon as the out-of-balance has been determined, the drive is switched off and the shaft uncoupled. A gripper arm then lifts the crankshaft and transfers it to the back of the machine for balance correction. At the same time a second gripper arm loads another shaft into position for testing, see Fig. 3. While the fresh crankshaft is being coupled up, the shaft that has been tested is brought automatically into position in relation to two drill heads, one vertical and one horizontal. The heads advance automatically and drill to the correct depths to remove the correct weights to bring the shaft into dynamic balance in accordance with the information passed from the control device. A shaft in the drilling position is shown in Fig. 4.

In the meantime, the measuring apparatus has already determined the figures for the shaft under test. Once again the gripper arms come into action. The balanced shaft is removed for transfer to the assembly line, the tested shaft is transferred to the drilling station and a fresh shaft is inserted for test. The complete sequence from loading for test to removal for transfer to the assembly line takes less than three minutes.

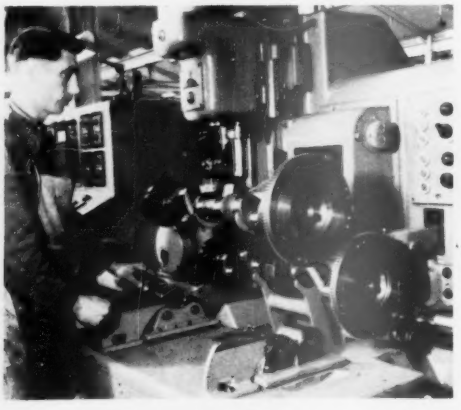


Fig. 3. The change-over of shafts

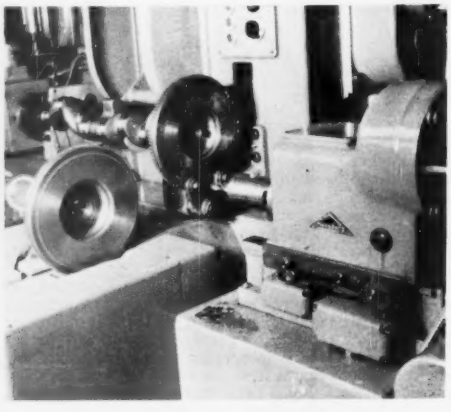


Fig. 4. Tested shaft in drilling position

QUALITY CONTROL

Savings Effected by an American Company Through the Introduction of Statistical Methods

MANUFACTURERS in the automotive, aircraft and other industries engaged in making precision mechanical products may find interest in a programme of quality production control which has saved one company $10,000,000 (£3,570,140) in four years. That is the record made by the International Harvester Company in twenty-one of its American plants by application of what it calls statistical quality control.

This form of control is, of course, essentially a means for tracing to their source and eliminating variations in the three major production factors—machines, materials and operators—that affect accuracy of component parts. No two job pieces produced are identically alike because the crystalline structure of the material varies minutely from piece to piece, cutting tools lose accuracy because of wear and heating by friction, and the least constant factor is the machine operator, whose working precision is affected from day to day by any of a number of causes. Among these causes, as manufacturers generally know, are changes in weather conditions, minor physical ailments, personal relations with other persons inside or outside the plant, and the worker's activities the night before his work shift.

Unlike customary final inspection of parts after a job run, the statistical quality control plan calls for frequent inspection measurement of a number of consecutive pieces as soon as they are machined. These readings are recorded on a chart close to the operator's machine. This visual record shows the operator at any time whether parts coming from his machine are within allowable tolerances. If they are, he is content; if outside, he knows something is wrong and corrects it promptly, if possible, by readjustment of tool or machine or calls on the toolroom for a resharpened tool. Thus, waste of material and labour time due to rejects of substandard parts is reduced to a minimum.

Control engineers use mathematical formulæ to envisage the pattern the variables may take and to establish the limits within which they should be kept. The application of statistics and the laws of probability to quality control can be demonstrated by throwing three dice. The most probable number to be rolled is 10 or 11; if some other total of spots shows most frequently, the dice must be loaded. In a production process the operator will try to keep his work at the set-up level but he does not become alarmed so long as the dots on his chart remain within the upper and lower control limits. If they do not, the chances are about 380 to 1 that there is a cause which must be traced to one of the variable factors and corrected, usually by a simple machine adjustment or tool change. Unless this is done before production continues, it will result in scrap.

This application of statistical analysis to quality control was suggested to the company's accounting department about fifteen years ago by Mr. Fowler McCormick, former chairman of the board of directors. So a group of young accountants was formed to take courses in mathematics and the laws of probability and to try to start a programme of statistical quality control in some of the plants. After World War II ended, the manufacturing research department was organized and the method has been improved and extended ever since. It is applied now in the company's steel mills, foundries, engine plants, tractor works, truck assembly lines, twine mills and repair depots. The system in some of the larger plants covers as many as 3,000 to 5,000 job operations, with up to 2,500 charts showing operators whether parts they are producing are within or outside of plus or minus tolerances.

The extent of success of the system in any plant or department depends upon the interest taken in it and the alertness to remedy trouble when it starts shown by the plant manager, production supervisors, tool engineers, machine operators and other members of the works teams. Key men in a number of plants were organized as action committees to find the cause of trouble when a chart showed a drop in work accuracy and to get the operation under control again as quickly as possible.

Each successive year the system has paid larger dividends through reduction of reject parts scrapped and material required to replace them, saving of labour cost that would have been invested in the rejected parts and salvage operations, and the production of parts to closer tolerances at lower

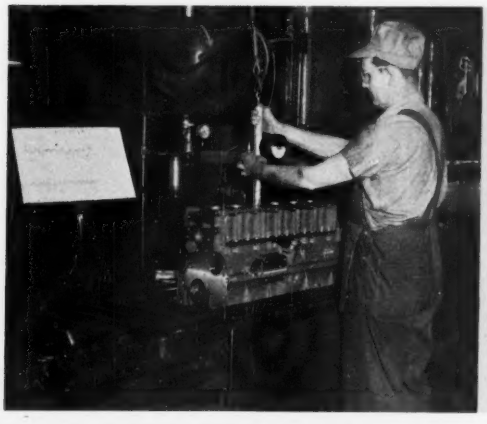


Fig. 1. Checking cylinder bores after finish honing. The control chart is on the left

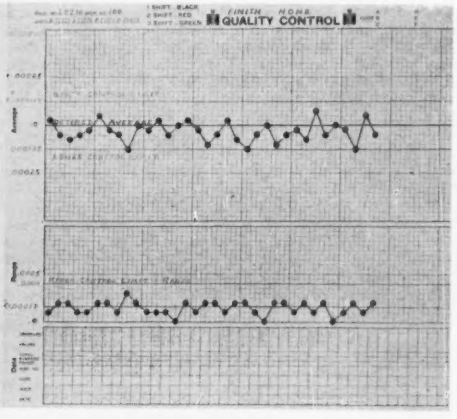


Fig. 2. A control chart for the operation shown in Fig. 1

cost. The saving of labour and material costs in 1951 was estimated to amount to $3,500,000 (£1,250,000). Some examples of gains made are as follows:

Scrap reduced 56 per cent in engine works foundry and machining operations, effecting a saving of $250,000 worth of labour and material.

More than 2,640 tons of steel saved in one year by the steel works through closer control of thickness dimensions of round and square bars and plates, thereby reducing waste of steel in machining operations.

Scrap volume cut 36 per cent in one foundry by control of mould and core hardness and pouring temperature.

Rework on crankcases reduced 47 per cent and scrap 56 per cent by control of machining operations.

Savings of $33,000 per year on grinding operations at one works by study of stock removal, grinding methods and use of new measuring equipment on gear hobs.

An improvement of 31·6 per cent in ratio of work expended on parts that had to be scrapped to total direct labour cost in all International Harvester plants during the last four years and an 11·7 per cent improvement in rework ratio.

These and many other savings and improvement of product were made with an increase of only 0·2 per cent in ratio of inspection cost to direct labour cost, although all costs of the programme have been charged to inspection.

Versatility of the control method is indicated by its application to diverse operations ranging from steel rod rolling and foundry casting to assembly and painting. Each specific type of job may require some variation in the technique, such as frequency of work inspection, ratio of pieces checked whether they are a consecutive number or chosen at random, and measures taken to trace imperfect product results most quickly to machine, material or operator. Details such as these have been dealt with in a series of papers published by the American Society for Product Control, 70 East 45th Street, New York 17. A few following examples of methods used at the I.H.C. Milwaukee engine works, however, contain information that provides answers to most of the questions likely to arise in the mind of a manufacturer who may wish to put a programme into effect.

The cylinders of engines selected at random after finish honing are checked for size (Fig. 1) and the average and range values are plotted on a chart (Fig. 2) which is mounted within view of the machine operator. It shows the actual inside diameter between upper and lower control limits with relation to the desired average and, on the lower chart, the range below the upper control limit. The checking needs to be done only three or four times daily because of past performance of the process.

In the foundry many control charts are posted where the workmen can see how well their performance comes within established limits. One of three charts records daily and by work shifts the percentage of defective barrel cores found by hourly checks in relation to the desired average and its upper control limit. On another chart is plotted the barrel core hardness, and the third chart shows the variation of specific gravity of the solution in core dipping tanks as obtained by a Baumé hydrometer. Control of the dip gravity within definite limits aids in obtaining a smooth core surface.

The foundry superintendent is kept posted on daily performance of the moulding units by charts set up in his office. Here shift performance is plotted each day together with the number of scrap castings compared with previous records. Control limits and averages are based on the gang's previous month's performance to indicate any significant improvement or deterioration in work done.

Difficulties encountered in hobbing transmission main shafts were remedied finally after installation of control charts which showed that, although the average size was fairly uniform, that of individual shafts varied considerably. After analysis and discussion of the charts, the machine feed and speed were increased, decreased and checked and different hobs used, all without satisfactory improvement in results. Putting the hob on another machine increased the uniformity somewhat but not enough. Further investigation disclosed that the operator had to shift the hob for each piece he hobbed and obtained only fifteen finished operations per hob. Also, friction burning of the hobs required grinding off .¼ in to resharpen them. Therefore the heat-treating method and the hardness of the shafts were investigated and a certain percentage were found to exceed the upper limit specification.

Holding the hardness on the low side helped to correct the trouble but without resulting in desired uniformity of product and sufficient decrease in tool cost. Then it was noted that hard spots showed on hobbed portions of the shafts, so the heat-treating process was refined. Considerable experimenting showed that shaft hardness could be lowered without impairing the function or the useful life of the shafts. After hardness specification limits were lowered, the operator needed to shift the hob only once for each three pieces and hobbed forty-five per hob. Moreover, only 0·01 in needed to be ground off dulled hobs.

Similar methods were followed in avoiding formation of an excess of heavy burrs when rough-cutting bevel gear teeth. These burrs had to be removed before making finish cuts or they would be dragged between the cutter and gear teeth and cause a scratched finish that often necessitated considerable reworking and sometimes resulted in scrap. Study of the tools used showed that they conformed with grinding specifications, and the gear metal was found by analysis to meet specifications although the Brinell hardness was somewhat low. From hardness readings taken on several lots of gear blanks at the normalizing operation over a period of time and through study of the results of rough-cutting the teeth, it was found that only a normal amount of burr was produced if the Brinell hardness ranged from 165 to 180.

Then followed experiments with cutting oils. The ratio between paraffin oil base and sulphur base was reduced in stages from 8:1 until it was established that a mixture of two parts paraffin base to one part sulphur base gave best results.

Later, when a change in the specification for gear material from 8600 steel to 4118 steel was made and teeth were rough cut, there were no traces of excess burrs and five job runs made in a two-month period showed that hardness of the 4118 steel was being held toward the high limit of specifications.

Statistical quality control, it is contended, results in more accurate and uniform duplicate parts and, by such savings as the foregoing, enables prices of assembled products to be kept competitively low. The method has been used to identify in batteries of machines those which were producing work below specification accuracy. Such machines were then either reconditioned or used for less exacting work. Furthermore, the inspection system has led to use of improved tools. In one example the replacement of a boring bar carrying two carbide-tipped tools with a two-step, carbide-tipped, inserted blade reamer made possible a 50 per cent reduction in the job tolerance range.

Not the least important benefit derived from the programme has been the attitude of shop employees towards the system. It is not a speed-up step by management that might be resented but is said to have won enthusiastic approval of operators because the charts at the machines aid them to do accurate work and blame for substandard work resulting in scrap often is traced to machines or material instead of to the worker.

Electrical Equipment

A LOOSE leaf catalogue setting out concisely the full range of Philips Industrial products is now available. The products are listed under subject headings for easy reference.

A revised arc welding catalogue, showing recent additions to Philips arc welding equipment, and an arc welding electrode wall chart are also now available. They may be obtained from Philips Electrical Ltd., Century House, Shaftesbury Avenue, London, W.C.2.

VEHICLE PERFORMANCE

A Method of Determining Drag, Engine Power Output, and Transmission Behaviour Under Operating Conditions

By Dipl. Ing. Z. Arkus-Duntov, M.E., S.A.E.

AT a certain stage of the development of the Allard J2X competition model a discrepancy between the considerable increase in engine power output on the dynamometer and the moderate improvement in road performance was suspected. Road-testing was confined to the M.I.R.A. proving grounds at Lindley, where high-speed facilities consist of straight timing strips and a perimeter track. Neither of these would permit the attainment of the maximum speeds of which the competition cars were capable and, while valuable for comparative testing of a well-established design, they did not furnish the precise information on which development could be based.

Since nowhere else in England could the high speeds be attained or continued for any length of time, the road tests were not conclusive and tended to raise more questions than they could settle. Tests which provide suspicions or hopes instead of positive information are likely to be misleading. According to the experience and character of the persons involved, a pessimistic or optimistic outlook for the future operational behaviour is assumed, with possible side-tracking of effort or unfulfilled expectations as a result.

The work at that stage was aimed at the development of a car which, within the imposed limitations, would be a strong contender at an impending Le Mans race. The engine, clutch, gearbox and brakes were under simultaneous development. The basic performance requirements were known and it was to determine at this stage how closely or how remotely the car performance related to these requirements. The modifications to the engine provided for 220-225 b.h.p. without endangering reliability and 260-270 b.h.p. with reliability unknown.

An engine delivering 205 b.h.p. at 4,500 r.p.m. under test-house conditions was installed in the chassis and the immediate questions were:—

1. Does the engine deliver its power in the chassis?

NOMENCLATURE

a	=	Acceleration
A	=	Mean acceleration
A_f	=	Frontal area
F	=	Force
F_e	=	Excess driving force
K	=	Drag factor
M_t	=	Total inertia
R	=	Resistance
R_a	=	Air resistance
R_r	=	Rolling resistance
ρ	=	Rolling resistance factor (actual equipment)
ρ_c	=	Rolling resistance factor (previously ascertained)
t	=	Mean time
t_1	=	Time on first trap
t_2	=	Time on second trap
V	=	Mean speed
V_1	=	Speed on first trap
V_2	=	Speed on second trap
ΔV	=	Speed difference
W	=	Total weight

2. What is the maximum speed?

Not before these questions were answered could further development work, if necessary, be undertaken. Full-scale wind tunnel tests, determination of rolling resistance, and high-speed runs on Continental roads could provide the required answers but, being outside the scope of existing limitations, had to be ruled out. The problem reduced itself to the establishment of a method of measuring resistance and power at highest attainable speeds, and of a method to calculate maximum speed from the highest attainable test conditions. It should be understood that information pertaining to engine behaviour gained at moderate speed is not necessarily applicable at high speeds. The turbulent or laminated atmosphere of varying pressure, in which the engine is breathing at high speed, not only affects engine performance in terms of density of the air it aspirates but, much more, by the freak pressures acting upon various carburettor orifices. It should also be remembered that conditions affecting the engine maximum output to the extent of, say, 10 per cent are of major importance. By losing 10 per cent of its power, a 200 h.p. car with maximum speed of 150 m.p.h. will not only be slower by some 5 to 7 m.p.h. but also will lose approximately 50 per cent of its acceleration in the 120-145 m.p.h. range. At high speed, after the tractive resistances have absorbed the bulk of

Test car at speed at the M.I.R.A. proving ground

power, a small percentage of total h.p. represents a large percentage of power available for acceleration. It may be relatively small in absolute terms but, nevertheless, it may enable the car to get ahead of a competing vehicle.

A remark by Mr. Dow, resident electrician at the M.I.R.A. proving ground, that should one set of timing lights fail, two-directional tests could still be carried out on a second set, provided the enlightenment that there were two successive timing traps on each strip. This gave a clue to the solution of the immediate problems.

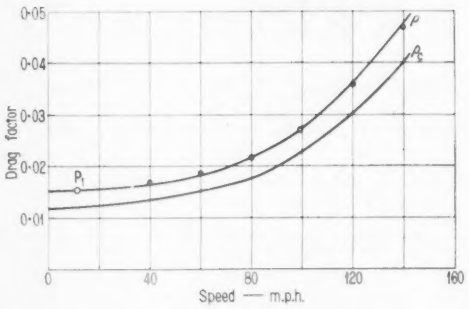


Fig. 1. Drag *v.* speed relationship

The timing strip consists of two runways, with grippy tarmac surface, each of about 1,800 yd total length divided into a run-up zone of 1,500 yd, two successive, independently-controlled, timing traps of 50 ft length each, and a braking zone of 300 yd. The traps are placed on opposite sides of the runways, allowing two-directional runs with resulting mean values independent of wind and ground inclination. The time is registered to 0·00001 sec and indicated separately for each successive trap.

When the car travels through the traps the electronic timing device furnishes two readings :—

Time consumed to travel the first 50 ft..... t_1

and

Time consumed to travel the succeeding 50 ft..... t_2

the corresponding speeds can be calculated :

$$\frac{50}{t_1} = V_1 \text{ ft/sec} = V_1 \times \frac{1}{1{\cdot}466} \text{ m.p.h.}$$

$$\frac{50}{t_2} = V_2 \text{ ft/sec} = V_2 \times \frac{1}{1{\cdot}466} \text{ m.p.h.}$$

also mean time :

$$\frac{t_1+t_2}{2} = t$$

mean speed :

$$\frac{V_1+V_2}{2} = V$$

speed difference :

$$\pm(V_1-V_2) = \Delta V$$

and acceleration :

$$\pm\frac{\Delta V}{t} = a$$

With this information, the state of motion of the car is positively defined and, with the total inertia M_t of the car known, the resultant force acting on the car can be determined. Choosing various modes of motion, some forces can be made preponderant and therefore determinable.

Designating Force as F and Resistance as R, the general equation of equilibrium for a body in motion is :

$$F = M_t \times a + \Sigma R \quad \ldots\ldots\ldots\ldots (1)$$

Should the driving force become zero and the motion continue, the equation of equilibrium is :

$$M_t \times a = \Sigma R \quad \ldots\ldots\ldots\ldots (2)$$

A run through the traps with the gear lever in neutral will correspond to equation (2). From this, with M_t known and a determined, ΣR for this particular speed can be calculated.

TABLE I. LOW SPEED COASTING RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	2·62546	12·97	2·92996	11·63	1·34	2·777	0·475
E	3·93067	8·67	4·15133	8·21	0·46	4·041	0·114
W	3·06824	11·10	3·60034	9·46	1·64	3·334	0·49
E	2·41722	14·12	2·49100	13·69	0·43	2·454	0·175

V = 11·2 m.p.h. = 16·29 ft/sec
A = 0·317 m.p.h/sec = 0·465 ft/sec²

TABLE II. HIGH SPEED COASTING RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·31694	107·5	0·31998	106·5	1·00	0·3181	3·14
E	0·32569	104·65	0·32911	103·6	1·05	0·327	3·21

V = 105·53 m.p.h. = 154·5 ft/sec
A = 3·175 m.p.h/sec = 4·65 ft/sec²

Dividing ΣR into its two major components, air resistance R_a and rolling resistance R_r, and knowing that

$$R_a = KA_fV^2 \quad \ldots\ldots\ldots\ldots (3)$$

K being the drag factor and A_f the frontal area, it can be assumed that R_a becomes negligibly small for very low values of V, and that the relationship

$$M_t \times a = R_r \quad \ldots\ldots\ldots\ldots (4)$$

from which R_r can be determined, will hold true for this condition.

A high speed coasting run corresponding to equation (2) will furnish the value of ΣR, and as an approximation it can be written

$$\Sigma R - R_r = R_a$$

Substituting the value of R_a in equation (3) and solving it for KA_f, the aerodynamic factor of the car is determined and R_a for any and all speeds can be calculated.

The drag characteristics are thus obtained and a power run through the traps in accordance with equation (1) will furnish the value of F_e = engine excess force at driving wheels.

Correlating speed and r.p.m., and making necessary correction for the mechanical efficiency, the actual power output in the chassis can be compared with the dynamometer output and corresponding conclusions drawn.

The practical application of the scheme outlined will be illustrated by the actual example, the test car being an Allard J2X of standard type.

Weight with driver, fuel and water as tested..... 2,615 lb

The inertia of rotating parts, tyres, wheels, brake-drums, etc. expressed as an equivalent additional weight ..... 160 lb

$$\text{Total mass} = \frac{2{,}800}{32{\cdot}17} = 86{\cdot}8 \text{ lb-ft/sec}^2$$

The car was equipped with new-gripped racing tyres, 600×16 front, 700×16 rear, at uniform pressure of 35 lb/in². The timing strip was dry.

A thorough warm-up of the whole car, including the gearbox, rear axle, and wheel bearings, is necessary. Half-a-dozen laps at speed on the perimeter track are usually sufficient to attain working temperatures and stability of readings. Should each succeeding run through the traps furnish progressively lower deceleration values, it is an indication that the stability condition is not yet attained and additional warm-up is advisable.

Any number of runs can be made and mean values used but two complete two-directional runs are usually sufficient. To minimize the effect of surface undulation, the two-directional runs for slow-speed range were confined to one set of traps only. This can be made possible by crossing the leads of the electronic timing apparatus.

Table I gives the results of low-speed range coasting runs. The values of t_1 and t_2 are furnished by the electronic timing device, the values of V_1 and V_2 can be read from the available table or calculated, the values of ΔV, t, and a

are calculated. Table II contains the results of high-speed range coasting runs, obtained in the manner described earlier.

With values of low-speed a and high-speed a available and the assumption that R_r is substantially constant over the speed range, the magnitude of R_r and R_a at tested speed can be calculated, KA_f determined, and characteristic curves giving the total or partial drag at any speed can be drawn.

The assumption of constancy of R_r is, however, inadmissible for the higher-speed range, and determination of R_r is somewhat complicated. The recommended procedure is to obtain the drag-speed relationship of the tyre used. The author was unable to obtain such information and established the R_r curve on the basis of slow-speed tests and known drag-speed characteristics of a similar tyre. The curve ρ_c in Fig 1 represents the drag-speed relationship of a particular tyre at 35 lb in^2 inflation pressure on a concrete surface.

Taking the value of a from Table I and the value of M_t, the equation (2) can be solved for low-speed rolling resistance :

$$R_r = 86{\cdot}8 \times 0{\cdot}465 = 40{\cdot}4 \text{ lb}$$

at 11·2 m.p.h.

The rolling resistance factor can be calculated from the relationship :

$$R_r = \rho W$$

$$\rho = \frac{40{\cdot}4}{2615} = 0{\cdot}0155$$

The value of ρ plotted in Fig 1 at corresponding speed (point p_1) lies higher than the corresponding value of the curve ρ_c.

The difference can be attributed mainly to the higher friction between the new-gripped tyre and tarmac surface, and to some amount of oil and oil-seal drag in the transmission elements. Making certain assumptions on the basis that rolling resistance is composed of speed-dependent and speed-independent factors, a relationship

$$\rho = \rho_c \times 1{\cdot}16 + 0{\cdot}0013$$

was evolved on the basis of which the curve ρ in Fig 1 was established. The validity of the ρ curve was rechecked by other analytical methods, and the author satisfied himself that the errors of assumption were in an order of magnitude that would not impair the practicability of the method. Believing that the drag-speed relationship can normally be obtained from the tyre manufacturer, description of the elaborate analytical methods employed is not included.

Applying the equation

$$R_r = \rho W \text{ lb}$$

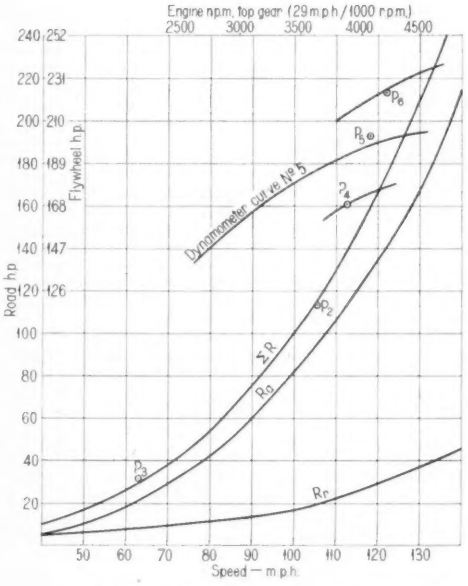


Fig. 2. Drag and road h.p. of test car

and

$$R_r = \frac{\rho W \times V}{550} \text{ h.p.}$$

the rolling resistance can be calculated for different speeds and is plotted in Fig 2.

Taking the value of a obtained in the high-speed test and solving the equation (2) we obtain :

$$86{\cdot}8 \times 4{\cdot}65 = 403 \text{ lb}$$

or $$\frac{403 \times 154{\cdot}5}{550} = 113 \text{ h.p.}$$

which represents the value of the total resistance of the car at a speed of 105·53 m.p.h.

The point designated by p_2 represents this value in Fig 2. The value of R_r at 105·53 m.p.h. read from the R_r curve represents 20 h.p. or

$$\frac{20 \times 550}{154{\cdot}5} = 71{\cdot}2 \text{ lb}$$

The air resistance at 105·53 m.p.h. is consequently

$R_a = 403 - 71{\cdot}2 = 331{\cdot}8$ lb

Solving equation (3) for KA_f, the aerodynamic characteristic of the car is obtained :

$$KA_f = \frac{331{\cdot}8}{154{\cdot}5^2} = 0{\cdot}0139$$

The air resistance curve calculated on the basis of $KA_f = 0{\cdot}0139$ is plotted in Fig 2 as well as the total resistance curve ΣR. With R_r, R_a and ΣR plotted over the speed range the drag characteristics of the car are positively defined.

It is advisable at this stage to check the correctness of calculated curves by experimentally determining ΣR at some intermediate speed. Table III contains the result of such intermediate coasting runs.

Substituting the values in equation (2) the total resistance is :

$$\Sigma R = 86{\cdot}8 \times 2{\cdot}18 = 189 \text{ lb}$$

or

$$\frac{189 \times 92}{550} = 31{\cdot}5 \text{ h.p.}$$

at 62·92 m.p.h.

The point (p_3) plotted on the graph, Fig 2, almost coincides with the curve of ΣR. The error is roughly 0·5 h.p. or some 1·5 per cent.

With resistance established, the full-throttle power runs were made in order to establish the actual engine output and maximum attainable speed. The results of the runs are contained in Table IV. The values of t_2 being smaller than the values of t_1 the acceleration is positive and excess of power can be calculated :

$$\frac{86{\cdot}8 \times 0{\cdot}802 \times 165}{550} = 21 \text{ h.p.}$$

at 112·8 m.p.h.

Taking the value of ΣR for this speed from the graph, Fig 2, the total power transmitted to the road is 161 h.p.

Plotting the value on the graph, Fig 2, point p_4, and tracing the semblance of a power curve, a maximum possible speed of barely 120 m.p.h. is indicated.

TABLE III. MEDIUM SPEED COASTING RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·53905	63·22	0·54570	62·4	0·82	0·5420	1·51
E	0·53750	63·39	0·54368	62·6	0·79	0·5410	1·47

V = 62·9 m.p.h. = 92 ft sec
A = 1·49 m.p.h/sec = 2·18 ft sec^2

TABLE IV. FULL THROTTLE POWER RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·29750	114·5	0·29656	114·66	0·16	0·2970	0·539
E	0·30712	111·0	0·30650	111·17	0·17	0·30681	0·554

V = 112·83 m.p.h. = 165 ft/sec
A = 0·547 m.p.h/sec = 0·802 ft sec^2

TABLE V. FULL THROTTLE POWER RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·2881	118·35	0·2875	118·63	0·28	0·2878	0·972
E	0·2901	117·60	0·2896	117·8	0·20	0·2898	0·69

V = 118·1 m.p.h. = 173 ft sec
A = 0·83 m.p.h sec = 1·216 ft sec²

To enable the comparison between the flywheel power, as determined on the dynamometer, and engine performance in the chassis, shown on the graph, Fig 2, a correlation of road and flywheel power is necessary. This is accomplished by the introduction of the r.p.m. scale, which is related to speed through the driving ratio of 29 m.p.h. /1,000 r.p.m., and the flywheel h.p. scale which is established on the basis of 95 per cent chassis efficiency in the top gear. The value of 95 per cent was estimated on consideration that part of the transmission losses are concealed in the ρ curve. It is now possible to plot the dynamometer curve on the graph, Fig 2, and obtain the reply to the question of engine output in the chassis.

A part of dynamometer curve No. 5, of uncorrected h.p., is plotted on the graph, Fig 2. This was the curve of the engine used at the time, and it can be seen that against some 195 h.p. delivered on the dynamometer, only 169 flywheel h.p. is delivered in the chassis.

With the fact and magnitude of h.p. loss positively established, it is a relatively simple matter to re-establish the performance, and the results of the power run after some changes have been made, are contained in Table V. The excess h.p. at the recorded speed of 118·1 m.p.h. is :

$$\frac{86{\cdot}8 \times 1{\cdot}216 \times 173}{550} = 33{\cdot}1 \text{ h.p.}$$

The total h.p. plotted on the graph, Fig 2, point p_5, falls on the dynamometer curve No. 5. Again the result is too close to be beyond suspicion and the closeness must be attributed to the interference of errors and assumptions.

With the application thus demonstrated, the utility of the timing strip is by no means exhausted. Applying the same principle of power determination, many aspects of engine behaviour throughout the engine speed range and the vehicle speed range can be isolated and studied. A fair indication of the gearbox behaviour can also be obtained. As before, a specific example of further application will be given.

The graph, Fig 3, represents acceleration in the upper ratios of a four-speed gearbox, which also was under development and was incorporated in the test car. The final drive ratio was 29 m.p.h 1,000 r.p.m., third gear 25·4 m.p.h 1,000 r.p.m. and second gear 18.4 m.p.h /1,000 r.p.m.

TABLE VI. FULL THROTTLE POWER RUNS

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·28698	118·73	0·28614	119·07	0·34	0·28656	1·188
E	0·27379	124·48	0·27320	124·72	0·24	0·27348	0·879

V = 121·7 m.p.h. = 178 ft sec
A = 1.03 m.p.h sec = 1·51 ft sec²

The acceleration, expressed in m.p.h sec, is plotted over the m.p.h. scale and the acceleration figures were obtained from the full-throttle runs in three gears at varying speed. In order to save space, the t_1 and t_2 values and the calculations are not reproduced.

The information that can be gained from the graph is of two-fold nature. The precise performance characteristics make it possible to adapt and use the car more intelligently. In this instance, the shifting points are clearly indicated, 88 m.p.h. up-shift into third gear and approximately at 114 m.p.h. up-shift into high gear. The main value, however, lies in the fact that the road h.p. curve can be calculated from instantaneous accelerations and a clear picture of the engine behaviour and of the functioning of the transmission elements obtained.

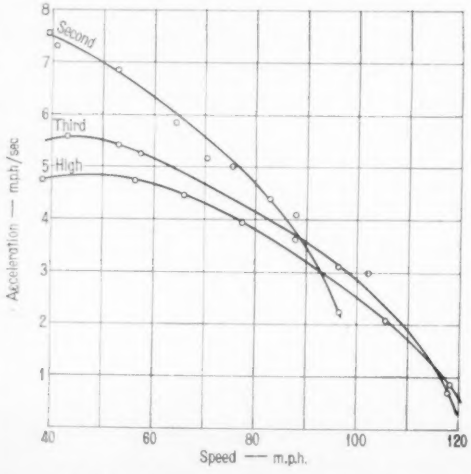


Fig. 3. Acceleration in three upper gear ratios

The graph, Fig 4, represents road h.p. curves calculated, as in the case of power runs, on the basis of equation :

$$\text{Road h.p.} = \frac{M_t \times a \times V}{550} + \Sigma R$$

Considering the high and the third-gear curves, it can be noted that their relationship is normal up to approximately 3,500 r.p.m. The third-gear curve lies a few h.p. below the high-gear curve, and indicates, apparently, the h.p. absorbed in the gearing. The mechanical efficiency of the third-gear ratio, if desired, can be calculated.

Above 3,500 r.p.m. the behaviour is obviously irregular. Since there is no source of power absorption that affects solely the high gear, the conclusion must be drawn that less power is developed or delivered in the high gear than in the third. Considering the second and third gear curves, it will be noted that power absorption in second gear is extremely high, indicating a somewhat disturbing condition in the gearbox. In this case also, at higher r.p.m. the power developed in second gear is superior to the output in third gear. From 4,600 r.p.m. upwards, notwithstanding the inordinate losses in the gearbox, the second-gear curve is well above the third-gear curve. There, again, the power is either not developed or not delivered and a third suspicion can be added that, since ΣR enters into the power calculations, the ΣR curve may be incorrect.

Regarding the car as a dynamometer moving at varying speed, with speed being the main variable, the h.p. curves are plotted in relation to speed on the graph, Fig 5. First the ΣR curve can be checked. Taking the 63 m.p.h. which was the checking point for the ΣR curve of the graph, Fig 2, 124 h.p. is delivered in high gear which corresponds to 2,170 r.p.m. Yet at 2,170 r.p.m. the third-gear curve shows 121 h.p., or 3 h.p. less, and the second-gear curve shows 114 h.p., or 10 h.p. less. It would seem reasonable to assume that these values are being absorbed in the gear ratios and it may, therefore, be concluded that there is nothing wrong with the ΣR curve at 63 m.p.h.

Considering at random a point in the higher speed range, it can be seen that at 90 m.p.h. 172 h.p. is delivered in high gear, which corresponds to 3,100 r.p.m. At this r.p.m. the third gear delivery is 166 h.p., which is still normal. Remembering that at 105 m.p.h. the ΣR value was measured and not calculated, it must be

assumed that the ΣR curve is definitely correct and consequently the irregularity in the h.p. delivery actually exists.

Since the discrepancy between the high and the third gear occurs under the same conditions of traction, the possibility of non-delivery of power must be excluded and it remains to infer that the variation in the airspeed is the cause of the irregularity in power development. The essential difference between experimentation for power plant analysis in one gear or in many, is the fact that the only major variable becomes the surrounding air stream, and one of the most elusive factors is then isolated and can be made the object of observation and experimentation.

Reverting to the actual example, it can be said that the moment the h.p. curves were plotted they confirmed the previous finding as far as the engine behaviour was concerned. They adequately explained, also, the rapid deterioration of the straight-cut gears in the gearbox. The 25 h.p. that was absorbed in second gear is equivalent to an heat input of some 1,060 B.Th.U. Applied for even a short time that is enough to soften the gears.

The time-saving value and positiveness of indication of such a test are of great value. Tested operationally on a short circuit, a gearbox as described might have passed undetected, with the result that on some other circuit, with higher overall ratio, and consequently longer uninterrupted use of the second gear, the apparently sound and tested gearbox would fail.

Table VII contains results of subsequent full throttle power runs, executed after some improvements had been made. The excess h.p. at 121·7 m.p.h. is :

$$\frac{86{\cdot}8\times1{\cdot}5\times17{\cdot}7}{550}=41{\cdot}6 \text{ h.p.}$$

Total h.p. delivered at this speed is :

172 + 41·6 = 213·6 road h.p.
(point p_6)

or

225 flywheel h.p. at 4,180 r.p.m.

The improvement over dynamometer performance of this particular engine can be attributed partly to the superiority of the cool Midland air as compared to the contaminated atmosphere of a testhouse in the London area, but mainly to the fact that is was found possible to turn some spoilers of performance into assets.

Plotting the point p_6 on the graph, Fig 2, and tracing the semblance of a power curve, a speed potential of approximately 132 m.p.h. was indicated. That value could not fulfil the requirements visualized. It was felt, keeping in mind the particular torque characteristic of the engine, that a maximum speed in the neighbourhood of 145 m.p.h. would be the sufficient but necessary minimum. Extrapolation of the curves in the graph, Fig 2, showed that the existing body shape would require some 280-290 h.p. to attain the estimated minimum of top speed. As mentioned previously, the output of roughly 260-270 h.p. could be obtained but, in the light of intermediate experiences, the reliability of the engine for a race of 24 hours duration would be questionable. The reduction of air resistance seemed to be a more rational course of action, and an enveloping body of simple form was decided upon. Mr. R. H. Canham, General Manager of Allard Motor Company, produced in a matter of a few weeks the body depicted in the illustration showing the author in action at the Le Mans race.

It was unfortunate that when the car was taken to the M.I.R.A. track, improvement work in progress was obstructing the timing strip. The hurried tests conducted at relatively low speed could not be considered reliable and, therefore, were treated with caution. No attempt could be made to establish excess forces. However, in view of establishing tentative gear ratios, the drag tests were carried out. The weight of the car was 3,000 lb and total mass was

$M_t = 98$ lb-ft/sec²

The results of high-speed runs are contain in Table VII.

Total resistance :

$$\Sigma R=\frac{98\times2{\cdot}28\times130}{550}=52{\cdot}8 \text{ h.p.}$$

at 88·6 mp.h.

Rolling resistance, corrected for increased weight :

$$R_r=14\times\frac{3000}{2615}=16 \text{ h.p. at } 88{\cdot}6 \text{ m.p.h.}$$

Air resistance :

$R_a = 52{\cdot}8 - 16 = 36{\cdot}8$ h.p. at 88·6 m.p.h

Aerodynamic factor :

$$KA_f=\frac{36{\cdot}8\times550}{130^3}=0{\cdot}0092$$

TABLE VII. HIGH SPEED COASTING RUNS, CAR WITH ENVELOPING BODY

Run	t_1	V_1	t_2	V_2	ΔV	t	a
W	0·38234	89·15	0·38531	88·47	0·68	0·38382	1·78
E	0·38491	88·6	0·38711	88·1	0·50	0·38601	1·30

V = 88·6 m.p.h. = 130 ft/sec
A = 1·54 m.p.h sec = 2·28 ft/sec²

An air resistance curve calculated with $KA_f = 0{\cdot}0092$ is plotted on the graph, Fig 6. Of the two rolling resistance curves, R_{r2} represents the curve of the test car corrected for increased weight, and R_{r1} a curve calculated on the basis of smooth concrete surface (which is the case of the Mulsanne straight at Le Mans), different tyre section and surface, and higher inflation pressure.

The resultant two ΣR curves represent the limits within which total resistance was expected to vary. The two cars were equipped with tentative ratios of 33·25 m.p.h 1,000 r.p.m. for car No. 4 and 32 m.p.h 1,000 r.p.m. for car No. 5. The power curves of both cars are plotted on the same graph. The intersections of the power curves with ΣR_1 and ΣR_2 show that the car No. 5 would have a slight advantage in top speed if ΣR_2 was correct, while No. 4 car would have the advantage if ΣR_1 was correct, due to the fact that 4,700 r.p.m. was the decided r.p.m. limit. However, should the calculation be incorrect and resistances exaggerated, No. 4 car reaching its safety r.p.m. would be some 5 m.p.h. faster.

As stated earlier, the tests were not regarded as conclusive, and the final

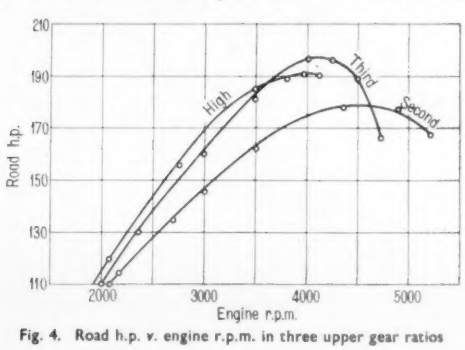


Fig. 4. Road h.p. v. engine r.p.m. in three upper gear ratios

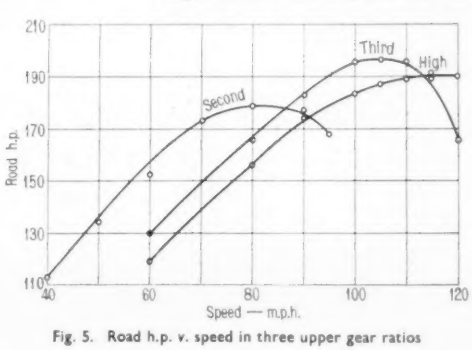


Fig. 5. Road h.p. v. speed in three upper gear ratios

decision as to the ratio to be used was left until the cars had completed practice laps of the race circuit.

Summarizing the applications and the methods described, it can be said that the state of motion and the instantaneous resulting forces acting upon a car can be determined with the help of the M.I.R.A. timing strip. The forces could be driving or retarding, and the author is of opinion that if brakes were the object of investigation instead of the power plant, a great deal of information could be gained. It is believed that this method, originally evolved as a substitute for wind-tunnel and high-speed track testing, has great merits on its own.

It must be emphasized that the work outlined was not the goal in itself, but represents a stage in the development programme of the time. The examples cited were not executed to demonstrate the utility of the method and, consequently, are not as demonstrative as they could well have been.

Reference to Le Mans as the reason for the development programme provokes interest in the actual performance of the cars in the race. It must be mentioned that the location of the spare wheel, in a well protruding from the lower run of the body on the near side, prohibited the use of an evolved exhaust system and resulted in some loss of power output.

The tentative ratios of 33·25 m.p.h. 1,000 r.p.m. for car No. 4 and 32 m.p.h. 1,000 r.p.m. for car No. 5 were installed. The crowded practice periods showed that both cars were reaching the necessary top speed and showed satisfactory behaviour, but the information obtained was summary and failed to provide positive indication as regards gear ratios. Accordingly, they were left unchanged. Car No. 4 was driven by Messrs. S. H. Allard and J. C. Fairman and car No. 5 by Mr. F. G. Curtis and the author.

Information gained during the race and pertaining to the subject treated was neither precise nor extensive, mainly due to the fact that heavy traffic in the first hours of the race did not provide the right setting for observation. Impaired brakes and darkness in the latter part of the race also did not contribute to scientific detachment. However, it was noted that 4,600 r.p.m. = 146·5 m.p.h. was obtained regularly, 4,700 r.p.m. = 150·5 m.p.h. on exceptionally clear runs, and 4,800 r.p.m. = 153·5 m.p.h. approximately in the slipstream of a faster car. These figures are based on revolution counter readings and pertain to car No. 5. Furthermore, in practice 5,500 r.p.m. = 140 m.p.h. was quickly attainable in third gear, though in the race only 4,500-4,700 r.p.m. was used in the gears. From car No. 4 is known that a maximum of 4,400 r.p.m. = 145·5 m.p.h. was regularly attained. The difference is less than 0·7 per cent and is well within possible reading or instrument errors.

In the race the cars showed similar lap speed. The figures and the general behaviour seem to indicate that car No. 4 was slightly overgeared and car No. 5 approximately correct, confirming the conclusions established at M.I.R.A. and represented on the graph, Fig 6. It is well to remember that circuit speed may be regarded as a chain reaction, and that the speed on the straight is very much affected by the speed on which the car leaves the proceeding bend. In a border-line gearing, faster speed through the bend and the addition of a couple of hundred r.p.m. in intermediate gears may affect the circuit speed appreciably. In the race,

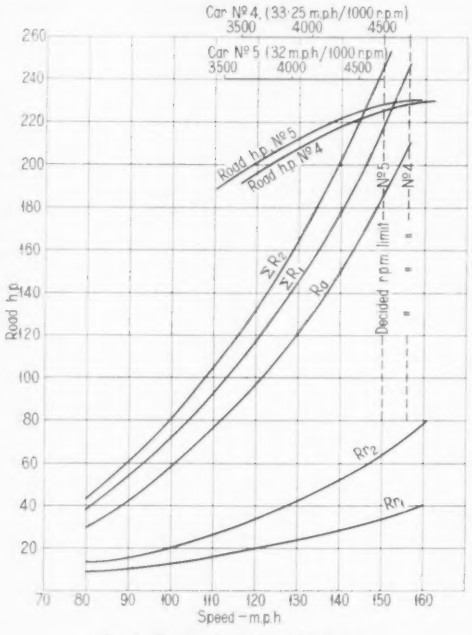
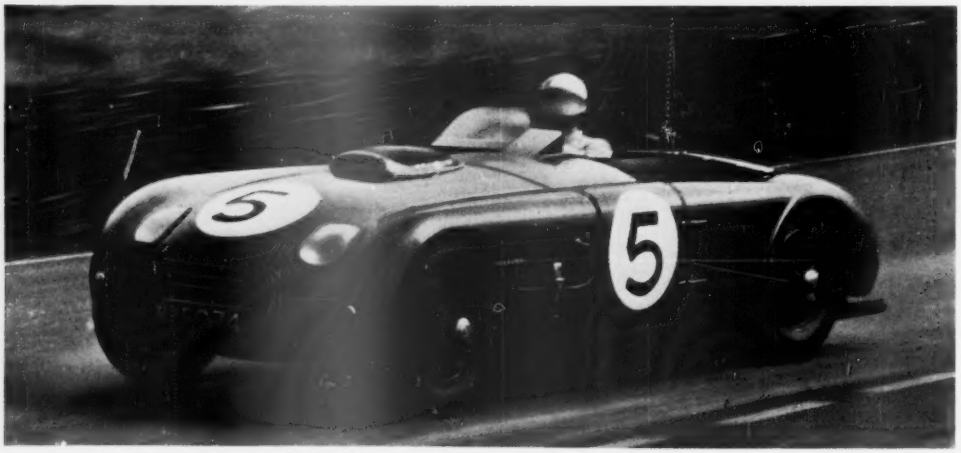


Fig. 6. Drag and road h.p. of race cars



Race car, with enveloping body, at speed on the Le Mans circuit

car No. 4 ran smoothly for thirteen hours, getting into 12th position in the third hour, 7th in the sixth hour, 5th in the thirteenth hour. Then, from causes not determined, the engine failed and the car was out of the race after fifteen hours.

Car No. 5, for all intents and purposes, was eliminated shortly before the end of the third hour. A loose shock absorber cut the hydraulic brake line on the approach to Mulsanne corner at approximately 140 m.p.h. Providentially, this potentially dangerous situation was resolved without injury to spectators or driver. At that time, No. 5 was the leading British car, lying 11th, showing over 98 m.p.h. average from the beginning of the race, lapping at 100 m.p.h. and able to lap at 104 m.p.h. if required. After resuming the race with impaired brakes, in 43rd position, it ran for a further twelve hours and was finally out of the race with a broken rear axle when lying in 17th position.

Acknowledgement

The author wishes to thank Mr. S. H. Allard for understanding support and personal participation in the development work which the author undertook during his association with the Allard Motor Company, and for permission to make public the figures. Also to Mr. R. G. Canham for the materialization of this development and to Messrs. Borg & Beck Co. Ltd., Wellworthy Ltd. (Al-Fin Division) and Zenith Carburetter Co. Ltd.

RECENT PUBLICATIONS

Brief Reviews of Current Technical Books

Metal Industry Handbook and Directory 1952

London: THE LOUIS CASSIER CO., LTD., Dorset House, Stamford Street, S.E.1. 1952. 6 × 8¾. 448 pp. Published in conjunction with the journal *Metal Industry* at a combined subscription of £2 12s. 0d.

The 41st edition of this handbook, well known to those engaged in the metal industry, whether buying, selling or manufacturing, contains the following revisions. The section "Properties of the Metals" has been brought up to date in order to clarify the present and future position as to supplies and prices. Each metal is treated individually from Aluminium to Zirconium and the 52 metals referred to include Cerium, Columbium, Gallium, Lanthanum, Osmium, Rhenium, Rubidium, Scandium and Thallium. The "Metals Statistics," covering the same metals, has been entirely rewritten. Official British specifications have been brought up to date and Proprietary Alloys is now revised. The Buyer's Directory is enlarged and facilitates the task of finding suppliers of plant for, and products of, the non-ferrous metals industry.

Layout follows previous issues and includes details concerning such diverse matters as Heat of Formation of Metallic Compounds, Tempering Metals, Solders and Fusible Alloys, and Specific Resistances of Metals and Alloys. There is a short section on Electroplating, Polishing and Finishing. Details of all metal trades associations and societies are given. The handbook will continue to provide interest and valuable information to all concerned with metals other than iron and steel.

The Sterling Area—An American Analysis

London: MUTUAL SECURITY AGENCY MISSION TO THE UNITED KINGDOM, Information Division, 1, Grosvenor Square, W.1. 1952. 8 × 10½. 672 pp. Price 21s.

This volume has been produced by the research staff of the Economic Co-operation Administration Mission to the United Kingdom, the predecessor of the present Mutual Security Agency Mission, because it was found, when they commenced to carry out their responsibilities in connection with the Marshall Plan, they needed a wealth of factual information in convenient form. This result of their work will be of inestimable value to all who have an interest in examining the economic problems faced by the Sterling Area countries after the war, and who wish to understand the nature of U.S. assistance.

It is emphasized that this study does not present a purely American point of view and that the sub-title "An American Analysis" only indicates that the contents are limited to subject matter of special interest to people in the U.S. The material was originally prepared in order to provide "a factual basis for the Mission's study of the United Kingdom's trading and financial situation and of the economic relations between the U.S. and other dollar countries on the one hand, and the U.K. and the sterling countries on the other."

Apart from being valuable to those interested in world economic problems and to business men who need to refer to the statistics, tables, charts and maps, the work is of wide general interest. It deals with almost every type of product imported and exported by the U.S., Great Britain and other sterling countries. Here will be found comparisons of post-war with pre-war business, right up to mid-1950, and the development of trade patterns generally, or by separate industries, can be followed.

The volume is divided into three main parts. The first deals with the composition, origins and growth of the Sterling Area, the second covers the countries or regions in the Sterling Area individually, and the third covers those commodities individually in which trade between the dollar countries and sterling countries is done.

Thus, the first part discusses the differing economies of the U.K. and overseas Sterling Area countries, the basic characteristics of the U.S. and Canada, the post-war importance of the "dollar gap," and the measures taken by the Sterling Area countries in dealing with it. The second section describes the economic background, foreign trade and balance of payments of all sterling countries with notes on their respective geography, history and governments. Part three covers grains, fats and oils, foods, beverages, tobacco, coal and chemicals, petroleum, iron and steel, non-ferrous minerals, machinery and vehicles, cotton and synthetic fibres, wool and other fibres, rubber and skins, and wood, pulp and paper.

Full details, in text and chart, are to be found of all trading in post-war years as compared with pre-war, for all the commodities referred to above and these are divided and sub-divided into readily referable information. Most of the statistical matter was obtained from various departments and agencies of the British Government and Governments of other Sterling Area countries and also from established trading organizations.

Who's Who in the Motor Industry

Edited by Roland C. Bellamy and Cyrus Andrews.

Grimsby: ROLAND C. BELLAMY PUBLICATIONS, Saint Mary's Gate. 1952. 5½ × 8½. 632 pp. Price 27s. 6d. post free.

Many individuals are secretly inquisitive about their fellow man, and especially when that man is someone in their own line of business. The success of a wide range of Who's Who books has been assured and this one covering the motor industry will follow suit. The reviewer is not certain if and when there has been a similar book but, since he has many connections with the industry, he has spent a long time already enjoying this book and it will find a permanent home in his office.

There are two main sections, the first listing firms connected with the industry together with directors' names, etc. It must be difficult to know where to draw the line here for all large garages and many smaller ones are included. The second section contains the "records" of some 2,500 individuals connected with the trade and details include their education, positions held, achievements, memberships of associations, hobbies and private addresses.

A smaller section covers the journals publishing news of or for the motor industry with "records" of their editors and motoring contributors. Details of various professional institutions and organizations connected with the industry are included.

We do not know how often it is intended to publish this book. Obviously, it will get out of date. This is the first edition and it is to be expected that such a book will have omissions. There are several alterations that could be made since the book went to press.

As for suggestions for future issues, one might well be to give more details concerning the firms contained in the first section. For example, a description of a firm as Accessory and Component Manufacturer could well be lengthened to state the accessories produced.

The biographical notes in the main section of the book are not skimped and this permanent record of one's associates in industry will be appreciated by many.

BEVEL GEAR CUTTING

Two Maag Developments for Accurate Work

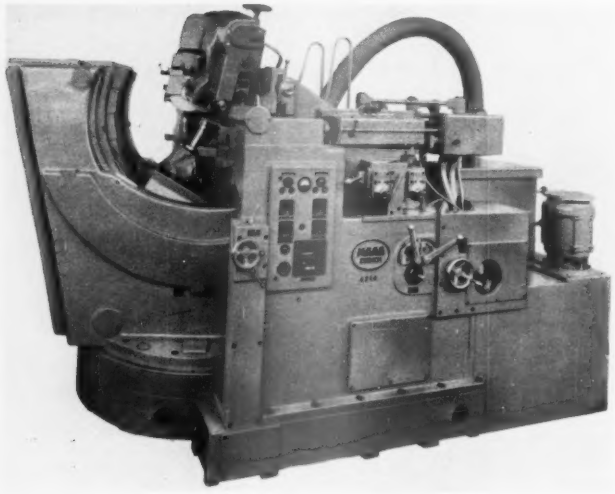


Fig. 1. KS42 Maag bevel gear grinding machine

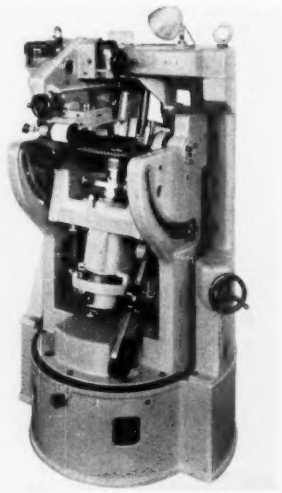


Fig. 2. KE42 bevel gear testing machine

A GEAR grinding and a gear testing machine recently developed by the Maag Gear Wheel Company, Zurich, Switzerland, are illustrated in Figs. 1 and 2. It is claimed that these new machines allow interchangeable gears to be produced with pitch, profile and cone-distance errors in the order of only 0.0001 in, and with a very high degree of surface finish. Subsequent matching or lapping of the gears is completely unnecessary.

The gear grinding machine operates on the generating principle. It will grind straight and oblique teeth of bevel gears with normal or corrected involute flanks up to a pitch diameter of 16·5 in and a ratio of 1:8. Two saucer-shaped wheels have a hydraulically controlled alternating motion along the tooth flanks, and a left and right hand tooth flank are ground simultaneously. Any wear of the grinding wheel of at least 0·0001 in is immediately compensated for by the well-known Maag compensating device.

Periodically, the outside diameter of the grinding wheel is automatically trued by means of a diamond dresser that is controlled by the compensating device. The lateral face of the grinding wheel is dressed by another diamond dresser swivelled by hand. As a result the grinding edge is kept sharp by removing only a very small amount of material from the wheels at each dressing.

The generating motion for producing the involute flank is executed by the workpiece. It is the result of a rotation of the shaft carrying the gear blank and a swivelling motion of the generating stand round the apex of the cone. These motions are performed through pitch blocks and steel bands. Indexing takes place after every single or double stroke of the grinding wheel heads, and is obtained by means of accurate index plates with hardened and ground teeth. Alteration to the ratios of the two motions governed is effected by changing the diameters of one or two pitch blocks.

The angular speed of the pitch block for the swivelling motion round the apex, that is, the feed motion, can be infinitely varied from 22 deg per minute to 460 deg per minute. Provision is also made for adjusting from 0 to 45 feet per minute, the speed of the slides carrying the grinding wheels. This allows the highest profile accuracy to be maintained. An automatic counting mechanism stops the machine when the required number of teeth have been ground.

But little heat is developed at the edge of the extremely sharp grinding wheels. Consequently, grinding is performed dry, the dust being removed by a device built into the machine. The grinding wheels, the indexing mechanism and the fan of the dust suction device are each driven by separate A.C. motors. If required, the machine can be supplied with a tip and root relief attachment, arranged to operate through a cam corresponding to the desired shape and amount of correction. This device is, however, only intended for large tooth corrections, since a small but generally sufficient tip and tooth relief, as well as a slight crowning, automatically results from the elasticity of the grinding wheel edge. As the essential machine components, such as the pitch blocks, the roll bands and the indexing discs are practically free from wear, and any wear of the grinding wheels is immediately and automatically compensated, extremely accurate bevel gears can be produced with almost perfect duplication.

As interchangeability of gears can be obtained only if it is possible to measure accurately the principal dimensions, the testing machine shown in Fig. 2 has been developed for use in conjunction with the gear grinding machine. It is designed to provide an absolute check on circular pitch, profile and lead angle. If desired, diagrams showing the continuity of the meshing line and the concentricity can be produced by means of the Maag centre-distance checking apparatus. B.S.A. Tools Limited are the agents for these machines.

Precision Barrelling

AMERICAN practice in the use of precision barrelling equipment is described in a folder of articles that can be obtained, free of charge, from Roto-Finish Ltd., 39, Park Street, London, W.1.

GLASS FIBRE IN PLASTICS*

Some Aspects of Its Use as a Reinforcing Material

By A. M. Dobson, B.Sc., A.Inst.P.

TWO forms of glass fibre are employed as the basic material from which reinforcing cloths and mat are made. They are: continuous filament, and staple fibre. Staple fibre consists of relatively short fibres, 6 to 15 in long. It is not widely employed in plastics applications, although some use has been found for it where a build-up of thickness is required between layers of continuous filament cloth. In addition, it has been combined with continuous filament yarn to give a cloth of increased hairiness, which can be used to advantage, for example, in rubber laminates to decrease the tendency to delaminate. All glass referred to throughout the remainder of this paper is of the continuous filament type. The properties of the filaments are given in the table.

Specific gravity	2·57
Thermal co-efficient of expansion	0·0000027/deg F
Thermal conductivity	7 B.Th.U ft² hr in deg F
Tensile strength	180,000 lb/in²
Modulus of elasticity	10×10^6

Use of cloth

Cloth is generally used where maximum strength is of greatest importance, or where the ease of application is a prime necessity. It is preferred because its uniform thickness makes it suitable for the production of thin sheets. Broadly speaking, three types of glass cloth are available as follows:—

a. Plain weave fabrics in which each warp and weft thread passes over one thread and under the next.
b. Satin weave fabrics, which may be designed as four, six or eight shaft, etc., in which each warp and weft yarn goes under one, and over three, five or seven thread yarns according to type. The eight shaft satins have been found most satisfactory in heavier fabrics. When laminated they have bi-directional properties substantially better than plain weave fabrics of comparable thickness.
c. Unidirectional fabrics are made with strong warp yarns and relatively few weaker weft yarns to give maximum strength in one direction.

Laminates with the highest overall mechanical properties are prepared from thin glass cloths, or at least include them in the outer skin. The strength of laminates decreases with increasing thickness of the cloth used, and this is most noticeable in compression strength.

These variations can be best understood by comparing the shape of the individual yarn in thin and thick fabrics, as well as plain and satin weaves. For example, in a plain fabric each yarn is crimped over and under other yarns, so that when under compression it behaves like a crimped column, and tends to collapse or bend further away from the neutral axis. Plastics reinforced with thin glass cloths are therefore stronger in compression, as the crimped effect is less pronounced. However, when a laminate receives a heavy impact, the larger bundles of crimped fibres tend to absorb the impact before the fibre column is fully stressed. Thus, laminates made from thick cloths give higher impact strengths than when thin cloths are used.

In order to obtain more nearly straight columns of fibre, the designer can use either the unidirectional fabrics or the eight shaft satin weave fabrics. In the first instance, the warp fibres are substantially straight, and all the crimp is in the relatively weak weft fibres. By laying these up with the strong warp yarn in the direction of maximum stresses, very high strengths can be obtained in the desired directions. The satin weave fabrics provide the strength of straight columns where the yarn floats over several cross yarns and the crimp only appears at intervals.

Plain weave fabrics are useful where design requirements call for:—

a. Uniformity of strength pattern in two directions at right angles.
b. Ease of removing air where contact or vacuum laminating techniques are used.
c. In the case of thicker cloths where considerable drawing properties are required.

Unidirectional fabrics are intended for use where:—

a. Maximum strength is required in one direction.
b. The load pattern can be predetermined, and the material laid up so as to meet the known requirements.
c. Local areas require additional stiffness or impact resistance.
d. Extreme strengths are required with a minimum weight.

Because these materials must be cross laminated to a specific pattern the cost of thei. brication in a laminate is increased by the hand labour involved.

Long shaft satin fabrics are best where:—

a. A part requiring high strength in all directions combined with light weight must be made with a minimum of labour for laying up.
b. High strength laminates can be achieved at low material cost by employing thick, long shaft satin in place of thin, square woven cloths.
c. Smooth surfaces and the decorative appearance characteristics of the long shaft satins can be used to advantage.

Cloth treatments

To produce a satisfactory bond between the glass and bonding agent it is necessary to remove the 2-3 per cent of size applied for bonding purposes during the manufacturing process. This can be effected by one of the following methods:—

a. *Caramelization.* This treatment is most suitable when the finished laminate is required for mechanical purposes. It consists of heating the cloth at about 300 deg C by hot plates, hot rolls or oven treatment until volatilization of the size ceases, and the cloth turns a golden brown.
b. *Full heat treatment.* This is usually employed where the final laminate is required for electrical purposes and, in particular, where further treatments are to be applied to the cloth. A two-stage heating cycle is carried out resulting in a cloth essentially free of size but having impaired abrasion resistance. As a result, special care is needed in handling the material.
c. *Washing process.* This is an aqueous washing treatment based on a spirit type soap.
d. *Other methods.* In America, the treatment known as 114 is used. This consists of removing all size by a full heat treatment, and then applying a compound that will key well both to the glass and to the laminating resin. It is known that the 114, or Volan treatment, is based on methacrylate chrome couples, and at various times it has been suggested that such compounds as vinytrichlorosilane would give similar properties.

In the States, there appear to be three main treatments: the O.C.F. 114, the 136 and the new Garan treatment. The application of one of these is essential if a laminate is to retain good physical and electrical properties under wet conditions.

In this country, no treatment based on a home manufactured product has been developed so far, although some work has been carried out on imported Volan. It is felt that this is one of the fields which the plastics and chemicals industries might well investigate, as it is important that such a treatment based on home manufactured materials, should become available in the very near future.

*A paper read at the 1952 Glass Fibre Reinforced Plastics Convention, organized by Fibreglass Ltd.

The construction of mats

In the past there have been available in this country two types of mat. The first is the diamond mat, which is constructed from filaments laid up in such a manner as to give a diamond pattern. This mat was developed for use where unidirectional properties were required from a reinforcing medium cheaper than a unidirectional woven cloth. It is bonded together with polyvinyl acetate, and apparently has been generally acceptable. One of the applications of this mat is in the production of boats by the Marco technique, where its open diamond pattern allows easy flow of the resin through the dry matrix when placed under suction.

The second type is the coiled mat, which is made from continuous filaments coiled in a random fashion. This type of mat has now been withdrawn from production in favour of the chopped strand mat which, at present, is made by showering 2 in filaments on to a conveyor, so as to give a random distribution. Here again, polyvinyl acetate is used to bond the mat together. This binder is probably quite suitable when a very low pressure, or none at all is being used, and when translucency is not an essential requirement of the final laminate. Because of the low softening point of polyvinyl acetate, the mat in this condition is probably unsuitable for matched metal moulding. Where deep draws are encountered the binder is likely to soften under the pressures and temperatures used, leading to tearing of the mat while the mould is closing. A continuous search is being made for other binders which can be applied preferably as a powder, but alternatively, as an emulsion. These must have one of the following two properties. Either good solubility in polyester resins, which will result in translucency, or a higher softening point so that the mat may be used in matched metal moulds.

Mats are generally used where the high strength obtainable with cloths is not required. In the U.S.A., probably the largest proportion of the output is employed for the production of translucent sheeting, and it has also been used to a wide extent in the production of boats, car bodies, as well as in the production by matched metal moulding of trays and boxes used, among other things, for carrying batteries.

Roving, and chopped fibres

In the dough moulding technique, $\frac{1}{8}$ in fibres are commonly used. On the other hand, for the flock pre-form technique, pre-cut 2 in chopped fibres were originally used in the United States, but to-day these have been almost entirely replaced by a roving, which is chopped *in situ*. All these types of material are still being supplied at present, but it is thought the future trend will be towards chopping *in situ* from roving.

Roving is currently made by winding a number of filaments parallel, that is, with no twist. Fibreglass Ltd. are developing a roving which will be used in the flock pre-form technique. The new roving is of the 60 end type. In this technique, the roving is fed through a cutter which chops it into 2 in lengths. The filaments then fall on to a grid on which they are held under suction.

Good bonding is essential in a prime quality roving, but during chopping it should break down into individual filaments. This break-down depends entirely upon the binders used, and by co-operation with the plastics industry, it is hoped to make further progress in the development of suitable binders.

Surface finish

It has been found that the surface finish of resin/glass fibre articles can be improved by the incorporation of a surfacing mat, which is a thin sheet material consisting of individual fibres of relatively large diameter, distributed at random. The binder at present used is a thermoset type, but alternatives are being investigated. This mat will hold a high percentage of resin on the surface, thus resulting in a good finish. Surface finish is also largely influenced by mineral fillers which may be incorporated, particularly in matched metal moulding.

The "Sulfinuz" Process

"SULFINUZATION," or sulphur cementation, is a process recently discovered in France for the impregnation with sulphur of ferrous surfaces, states G. de Lavalette, in the *S.I.A. Journal*, May, 1952. Following the treatment, which consists of a salt bath, the coefficient of friction of the surfaces treated is greatly reduced, resulting in a considerable improvement of wear resistance which is not accompanied by an increase in surface hardness. The treatment can be applied to ferrous metals of any composition, including stainless steels.

Two experiments described illustrate the properties of sulfinuzed surfaces. Two cylindrical test pieces of medium-hard steel were rotated between V-blocks under a load of 500 kg with fluid vaseline lubrication. The non-sulfinuzed test piece scuffed within a minute, whereas the sulfinuzed one retained a perfectly smooth surface throughout the whole experiment lasting 10 min, even after doubling and trebling the load. The second test was run completely dry and resulted in an almost instantaneous scuffing of the non-sulfinuzed piece. The sulfinuzed test piece, however, ran for 7 min, acquiring a temperature of about 500 deg C, without damage to the surface. Due to the high heat generated, however, its diameter was reduced from 6·35 to 5·5 mm, and its length increased by more than 6 mm. (*M.I.R.A. Abstract No.* 5942.)

Aluminium Radiators

INFORMATION gained with a first aluminium radiator was used to build a second aluminium radiator which satisfactorily completed 27,000 miles in a Cadillac car. In anticipation of a copper shortage, a limited field test was begun in 1950 with some 35 aluminium radiators.

J. R. Holmes, reporting in *S.A.E. S.P.* 85, Panel Disc, March 5, 1952, states: Experience to date shows that the cost of an aluminium radiator is much higher than that of a copper radiator, the salt bath used in brazing being very expensive. There is no known solder, compatible with the coolants, which can be used on aluminium, and the joints must be either brazed or welded. This, together with the fact that the core must be hand-welded, raises labour costs. In all brazing operations there are voids which must be welded, and some may not be accessible to a welding torch or arc.

Aluminium has a thermal conductivity about one-half that of copper but the cooling performance of aluminium radiators can be made comparable to that of copper radiators without excessive increase in gauge or in surface area. With the available aluminium brazing sheets, the tooling for the cellular-type core is practically impossible. Hence, effort is being concentrated on the tube and fin, and tube and centre core types. (*M.I.R.A. Abstract No.* 5923.)

Application of Cutting Oil

TOOL-LIFE tests were conducted with a single-point tool in a continuous turning operation under specified cutting conditions until failure of the tool point occurred. Six high-speed steel, single-point tools were ground to shape and in two of these two grooves were added in the side clearance face so that the "tips" of the grooves were within the first 0·001 in back from the nose of the tool.

Test results, given by W. E. Lauterbach in *Lubrication Engineering*, June, 1952, show that with the cutting oil directed down on the standard tool, an average tool life of 8·37 min was obtained. With the cutting oil directed upwards as described, and a standard tool, the average tool life was 17·35 min. With the grooved tool, the tool life was further lengthened to 18·98 min, a total increase of 127 per cent.

It is concluded that, in a turning operation, cutting oil performs its functions of providing lubrication between tool and chip and tool and workpiece, and of cooling the cutting tool, more effectively when the oil is directed up into the clearance crevice than when it is directed downwards over tool and chip. It appears that the oil has freer access to the actual cutting edge of the tool when introduced into the clearance crevice instead of being flooded over the tool from the top where the chip is being removed from the workpiece. (*M.I.R.A. Abstract No.* 5956.)

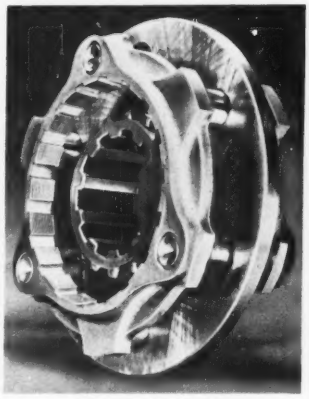
SPLIT PIN SYNCHRONIZERS

New American Units for Heavy-duty Gearboxes

A NEW design of synchronizer unit for the gearboxes of heavy-duty motor vehicles has recently been developed by Clark Equipment Co., Buchanan, Michigan, U.S.A. It is stated that this unit, the "split pin" synchronizer, is an improvement on the more conventional types of mechanism. Briefly, the advantages claimed are: It is of simpler construction, more dependable, more compact and has longer life. The unit comprises a steel hub with a central flange, two bronze-alloy synchronizing cones, three "blocker stay" pins, three energizer "split pins" and six small leaf springs. It must be explained that the term "split pin" is not used in the usual sense, but is applied to a cylindrical pin that is split on its longitudinal axis into completely separate halves.

The hub bore is splined to match the mainshaft, and the engagement teeth are cut in its outer periphery. There are six chamfered holes drilled in the central flange of the hub. Blocker pins pass through three alternate holes, and split pins through the others. In a typical unit, the blocker pins will be ½ in diameter, and the holes will be a little larger to allow the pins to pass freely through. The blocker pins are shouldered at each end, and an annular groove is machined midway between the shoulders. This groove is ⅛ in deep, and has sides at an angle of 57 deg in relation to the longitudinal axis of the pin. The synchronizer cone rings, one on each side of the hub flange, are assembled on to the reduced ends of the blocker pins, which are finally peened to secure the assembly. To break down the oil film and give a smooth disengaging action, the internal faces of the cones are divided into segments and grooved.

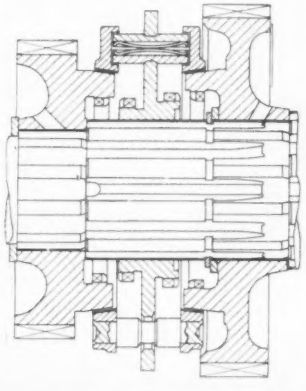
An annular groove is machined in the periphery of each split pin midway between the ends. It is ⅛ in deep, but the sides are at an angle of 30 deg instead of at 57 deg as on the blocker pin. The pin is then split along its longitudinal axis into two segments, each rather less than semi-circular, and a rectangular groove is machined in the flat face of each. When the segments are placed together the grooves form a square hole along the principal axis. Two small leaf springs are assembled back to back in the grooves. They can be compressed sufficiently to allow the assembled split pin to be passed through the appropriate hole in the hub flange. Each end of the split pin registers in a blind hole on the inside face of a cone ring. When the assembly is completed the leaf springs



Simplicity is the outstanding feature of the split pin synchronizer

force the segments apart until the annular groove makes contact with the wall of the hole in the hub flange. Therefore, in the completed assembly the only effective contact between the hub and the cones is through the energizer split pins.

The action of the unit is quite simple. At the beginning of the shift motion, the hub is moved on its splines towards the main-shaft gear. Because of the spring pressure between the split pins and the hub flange, the synchronizer cones are carried forward until light contact is made with the conical periphery of the appropriate gear. This light engagement effects an angular displacement of the cone in relation to the hub. Only a slight rotation of the cone brings the annular grooves on the blocker pins into contact with the walls of the holes in the hub plate, and provides an effective lock against any further relative axial movement between the hub and the cones. Under these conditions, the force applied to effect the gear shift can only engage the cones more firmly until such time as the hub and the gear rotate at the same speed. Then the torque holding the blocker pin grooves in contact with the holes in the flange is automatically relieved, and the shift can be completed.



The cone rings are mounted on pins which are passed through holes in the hub flange

As the shift proceeds, the hub is moved towards one end of the split pins, compressing the springs at that end, while the segments at the other end are forced further apart. In effect, the split pin becomes tapered, and the spring forces are such as to tend to move the pins in the opposite sense to the direction of shift. Thus, as soon as the shift is completed, the spring forces cause the split pins, and with them the assembly of cones and blocker pins, to move in the opposite direction to the shift. This movement is sufficient to bring the appropriate cone out of engagement so that it may rotate freely with clearance. It is stated that with this synchronizing unit, the shift is completed quickly, quietly and without any damage to the engaging parts.

Toothless Bandsaw

AS a consequence of an excessively high expenditure on bandsaw blades for sawing stainless steel, Solar Aircraft Co. of San Diego, California, conducted an investigation of sawing procedures aimed to reduce costs. According to a report in *Materials and Methods*, July, 1952, it was noted that the back of the blade appeared to cut as well as the toothed edge. Following up this phenomenon, a length of ordinary steel box-strapping was tried as a saw. It was found to operate satisfactorily, to be better than a conventional blade, and to be safer in use.

The Company has made only preliminary studies of this technique, but has found that toothless blades last much longer than ordinary bandsaw blades. Of less expensive material more readily in supply, the toothless blades can cut stainless or other high alloy steels, and have satisfactorily cut tool bits, files, and cobalt steel. They operate best at high speeds and have been employed successfully only on bandsaws. (2031)

RADIANT HEAT

Notes on an Efficient Method of Drying Paint

RADIANT heat as a means of baking paint surfaces is now widely used throughout industry. The rays in a radiant heat oven have a greater wavelength than the red rays at the end of the visible. As soon as the lamps in a radiant heat oven are switched on, energy in the form of rays is radiated in waves to the surface of the article to be treated. This energy is transformed into heat as soon as it reaches the product.

Much research has been carried out both on lamps and reflectors. For several years a chromium - plated reflector was the type normally used for radiant heating installations, but recent developments have shown that gold is a far more efficient reflector. The relative reflective efficiencies of various surfaces is:—

Gold	98 per cent
Processed aluminium	85 per cent
Polished aluminium	65-75 per cent
Chromium or nickel	60 per cent
Painted aluminium	40-50 per cent

In the production of gold reflectors, specular gold is applied over intervening coats of copper and bright nickel on a steel base. This method prevents the gold from amalgamating with the base metal, thus giving a longer life for the reflector. Self-container reflector lamps and gold oven surfaces are used in some applications.

In this country radiant heat lamps have been developed by Osram with a loading of 250 W. They are operated at 115 volts. This lower voltage allows a more robust lamp to be produced than would be possible for a normal 230-250 V circuit. The lamp filament is made of tungsten. Trough reflectors containing several lamps have gradually replaced the former parabolic reflectors. For a complete oven, these reflectors are built up into banks.

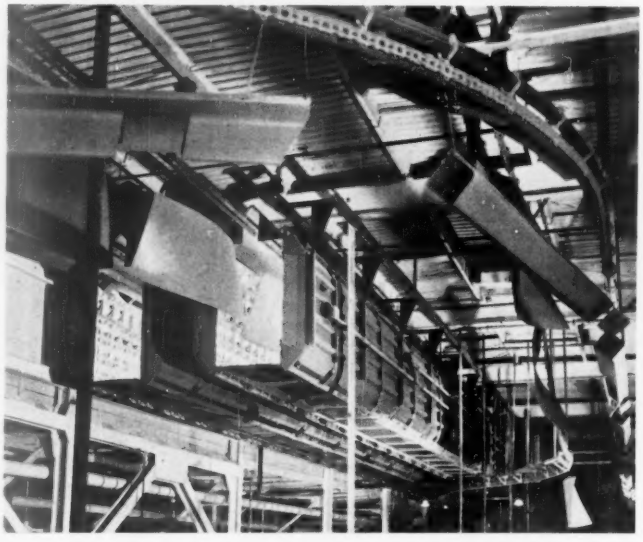


Fig. 1. Two radiant heat ovens suspended from the ceiling to conserve floor space

Radiant heat has many advantages. It brings the product to the required heat more quickly than any other method and therefore shortens the cycle time and speeds up output. As it is a source of instant heat, it can also be economical. For example, the plant can be shut down during the lunch hour break without any disadvantage, since when it is switched on again there is no time lag waiting for the oven to come to the specified temperature. Furthermore, if the lamp arrangements are correct, uniform heat is assured over the whole surface of the work, regardless of variations in cross-section.

Some American installations of radiant heat equipment are shown in the accompanying illustrations. Fig. 1 shows two 75 ft, ceiling-suspended ovens operating at White's Motor Coach Division, Ohio. These units operate in conjunction with a four-stage spray-type parts cleaning machine, an air blast dry-off oven, two electrostatic paint spray booths and two infra-red ovens for drying a primer coat. The parts are moved in sequence on an electrically operated conveyor. Ceiling mounting has been used for this operation to save floor space. It is situated 30 ft above the floor, has an output capacity of 540 kW and is series-parallel connected to operate off a 440 volts supply. Fig. 2 shows an installation at the Studebaker-Hamilton factory in Canada. In this oven the gloss enamel on a car body is baked in a through time of 24 minutes.

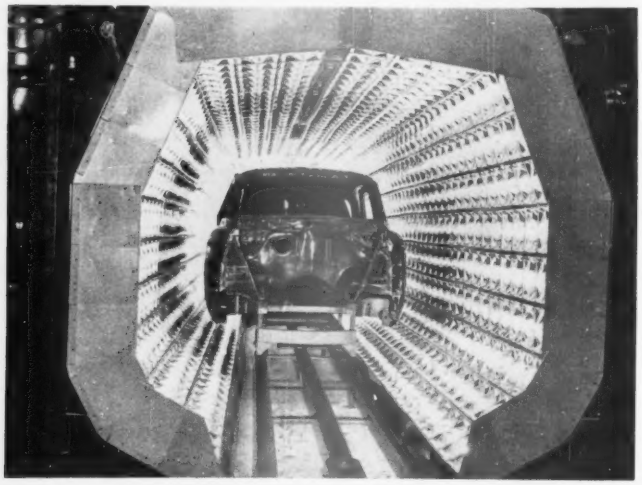


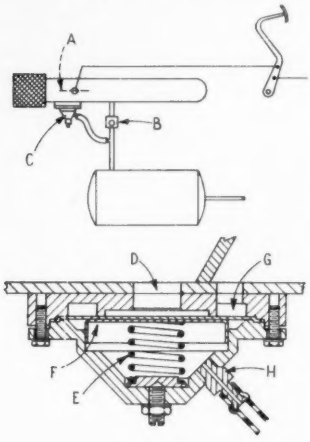
Fig. 2. An installation for baking gloss enamel on a car body

CURRENT PATENTS

A Comprehensive Review of Recent Automobile Specifications

Operation of auxiliaries by manifold depression

ON diesel-engines vehicles, the brakes, doors, windscreen wipers and other auxiliary equipment may be operated by the depression in the air manifold without the use of a special pump. The manifold is fitted with a throttle and a by-pass around the throttle is controlled by a valve device influenced by a spring and the reduction in pressure. Hitherto, the spring tended to hold the valve open until the throttle was closed and the reduction of pressure behind the throttle



No. 671132

effected a closure. With that arrangement the air flow to the cylinders will be temporarily throttled, or even shut off, so that incomplete combustion may occur or the engine may stop.

By this invention the valve is held by its spring in the closed position and communication with the atmosphere is effected by means of the by-pass whenever a predetermined reduction in pressure occurs in the manifold, in the operating chamber of the auxiliary or in the air reservoir.

Shown diagrammatically, the brake pedal is linked to the throttle valve A and also to the brake operating valve (not shown). The air manifold is connected by a conduit, controlled by a non-return valve B, to the air reservoir of a vacuum brake installation. A control valve C in the wall of the manifold upstream of the throttle valve encloses a chamber connected by a tube to the conduit behind the non-return valve.

In the control valve a diaphragm is pressed against a seating, around passage D opening to the manifold, by spring E and a guided metal cap F. The diaphragm also closes an annular chamber G communicating with the manifold downstream of the throttle valve.

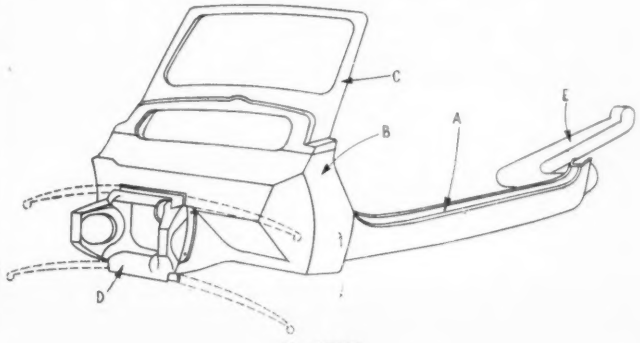
When the brakes are applied, the throttle valve is closed and the engine draws air from the reservoir, reducing the pressure therein. This lowering of pressure is communicated to the underside of the diaphragm and to the annular chamber G. Atmospheric pressure upstream of the throttle valve, however, unbalances the diaphragm and the throttle valve is by-passed by way of passage D and chamber G. When the throttle valve is shut the flow of air is only interrupted momentarily and consequently the engine will not smoke or falter. In order that operation of the auxiliaries is not endangered by damage to or faulty operation of the diaphragm, a fine-bore nipple H serves as a conduit of restricted cross section to establish a balance of pressure. *Patent No.* 671132. *Robert Bosch G.M.B.H. (Germany).*

Lightweight body-chassis construction

THE use of light alloy castings for the stress-bearing members of automobiles is by no means a modern development, but in this invention particular attention has been devoted to the problems associated with the manufacture of such constructions. From the illustration it will be seen that the major components that are to form the basis of the vehicle comprise the left-hand and right-hand longerons A, which could also constitute the side panels, and a bulkhead structure which provides the main crossmember. To facilitate manufacture, these main units are built up from a number of smaller castings.

For example, the dashboard or bulkhead unit is composed of a lower casting B which provides the side scuttle panels integrally with the toe-board and lower dash panel details. Above this, a second casting C embraces the upper screen pillars, the front top rail and the scuttle top panel. A third casting D forms a central cradle which, when attached to the lower casting B, is capable of accommodating the mountings of the engine and suspension units.

It is not considered necessary for the main side members to extend forward of the dashboard unit and, similarly, at the rearward end the side members terminate in front of the rear axle which is carried by cast members E spigot-mounted to the end of the longerons. *Patent No.* 670070. *J. A. Gregoire (France).*

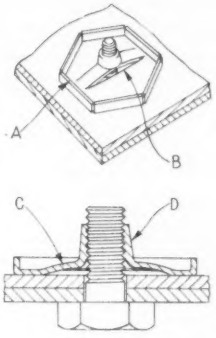


No. 670070

Sheet metal, self-locking nut

THIS self-locking nut has a tubular, screw-threaded hub formed on a bowed strip integral with the main base. Formed by slitting, punching and bending operations, the nut has a hexagon base having upturned peripheral flanges A to provide tool-engageable surfaces. Two slightly curved slits B define the strip C which is simultaneously formed with the hub D and bowed from the base. The hub is tapped with a parallel thread.

As initially applied, the nut threads easily on the bolt, but when tightened on



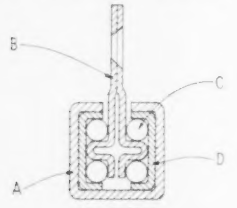
No. 672150

the work the strip is highly constrained and exerts a constant axial thrust on the threads. Furthermore, in the final tightening of the nut, the hub is slightly distorted from its cylindrical form and this deformation creates a binding engagement of the threads giving a positive and permanent lock. *Patent No.* 672150, *Tinnerman Products Inc. (U.S.A.).*

Seat slides

RUNNERS to give a sliding action are common components of both private and commercial bodywork, and this invention has the object of providing runners that will both carry the load and take care of the reaction without tending to jam or stick. Primarily intended for seat operation, but applicable to other body items that require a sliding action, this device consists of an outer channel member A which has its side flanges inturned at the top to provide a longitudinal slot. In the channel is symmetrically disposed an inner member B of cruciform section and having a flange extending through the slot in the outer member.

An essential feature of this inner cruciform member is that it is fabricated of spring steel or similar resilient material, and in each of the angles formed by the cruciform section a ball C is introduced. Four such balls comprise a set, of which



No. 669664

there may be two or more in the complete length of slide, and since the inner slide may be closed together against its natural resilience before the insertion of the balls, the tendency is for the balls to remain in close contact with the walls of the two slide members.

If considered desirable, a cage may be introduced to retain the balls and prevent longitudinal movement. It is also possible to give only limited longitudinal movement by introducing a second channel D with flanged ends. *Patent No.* 669664, *Cox and Co. (Watford) Ltd., and C. Toon.*

Folding occasional seats

OCCASIONAL seats that face the direction of travel are a common feature of limousine coachwork, and are frequently found in large saloon bodies. If the collapsible seat is mounted on a board that, when raised, forms a part of the back face of the division or seat back, it is not always possible to accommodate a full-depth hinged board in the limited space between the division and the rear seat heel board. In such cases it is usual to arrange a shallow hinged panel to fill the division recess after full allowance has been made for the largest possible hinged board.

In the invention the division recess is not completely closed, but the space remaining after the seat board has been hinged up is divided from the portion occupied by the folded seat. A shelf which is integral with the seat mechanism is so arranged that it falls into place between the side walls of the recess when the seat is stowed. The occasional seat is of the type that is erected after the division panel, which carries it, has been lowered, and it will be seen that the shelf board A is attached to the lower extremities of the seat back arms B by bracket arms C. Since the shelf also provides the means of securing the seat in its closed position, retaining catches D are fitted to each side the recess, and these catches engage with similar members E on the underside of the shelf. To enable these catches to interlock the shelf must be capable of movement relative to the seat back, and to achieve this the bracket arms C are pivoted at F, but the range of movement is limited by a pin-and-slot device G.

To lower the division panel, the shelf board is tilted forwards thus disengaging the catches, and by means of a fillet on its rearward edge, the panel is hinged downwards to rest on the floor when the seat can be erected by pulling the seat back. When the seat is in use the shelf board is held in a horizontal position beneath the rear edge of the cushion where it does not restrict the leg-room of the rear seat occupants. *Patent No.* 669687. *Austin Motor Co. Ltd.*

Flame-cutting of resistant metals

IT is known that metal bodies which are not readily amenable to ordinary flame machining methods may be flame-machined by using an adjuvant powder in conjunction with a preheating flame and a metal-removing oxygen jet. Adjuvant powders are iron (in the form of cast iron or steel), ferro-manganese, manganese, and iron-tin alloys. Powdered aluminium has been proposed both as a means for improving the efficiency of the flame-cutting process in general, and in particular for flame-cutting cast iron. Before the present invention iron had proved to be the best for thermo-chemically removing metal from both ferrous and non-ferrous metals and alloys.

It is claimed for this invention that by using an adjuvant consisting of from 10 to 65 per cent powdered aluminium and the balance of powdered iron, improved thermo-chemical removal of metal can be obtained. Relatively great thicknesses of non-ferrous metals can be flame-cut and flame-scarfed, without the necessity of a separate pre-heating operation, at a high rate and producing surfaces of relatively good quality.

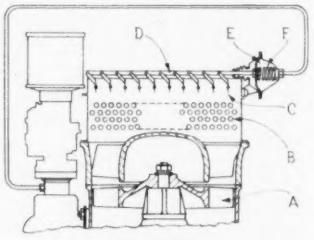
Particularly good results are obtained in cutting, it is said, when the powder is suspended in a gas such as air and blown through a separate tube having its outlet adjacent to the tip of the flame nozzle so that the powder penetrates the envelope of the preheating flame and is heated to a high temperature before merging with the oxygen jet. In scarfing, good results are obtained by suspending the powder in the oxygen jet before discharge from the nozzle so that the stream of powder and oxygen actually issue thoroughly mixed with each other.

A slab of nickel 15·2 cm thick and 76·2 cm wide was successfully flame-cut using a mixture of iron and aluminium wherein the aluminium constituted 33 per cent by weight. *Patent No.* 671863. *Linde Air Products Co. (U.S.A.).*

Control of fan-cooled engines

WITH an unregulated admission of air to a fan-cooled engine, difficulties may be experienced due to overcooling in cold weather, when driving downhill, or when running in congested traffic areas. It is proposed, therefore, to provide an adjustable shutter in the air intake which is automatically actuated in response to manifold depression.

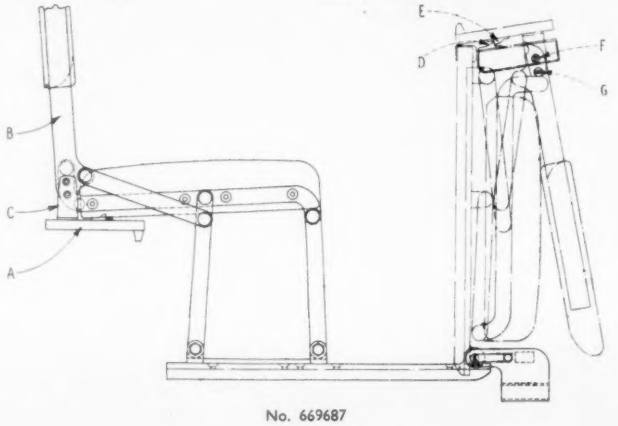
In the example, an engine-driven fan A draws in cooling air through a suction casing in which is housed an oil cooler B.



No. 672369

The cross-sectional area of the intake is varied by a shutter comprising pivotally mounted, parallel vanes C, constrained to the open position shown. An operating rod D, having an appropriate number of rigidly attached fingers engaging the vanes, is actuated by a diaphragm E influenced by the manifold depression.

With the engine running at normal speed with a wide open throttle, the loading spring F will hold rod D in the open position. At low engine speeds and consequently increased depression in the manifold the constraint of spring F is overcome and the vanes are moved to a partially or completely closed position. Additionally, a thermally responsive device may be fitted to adjust a stop limiting opening movement, and an interconnection with rod D may be provided to admit warm air to the carburettor when the engine is idling. *Patent No.* 672369. *Tatra, n.p. and J. Makerle (Czechoslovakia).*



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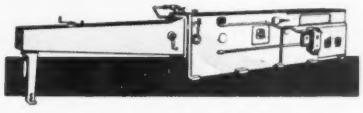

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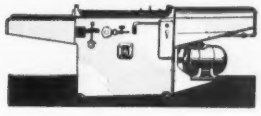

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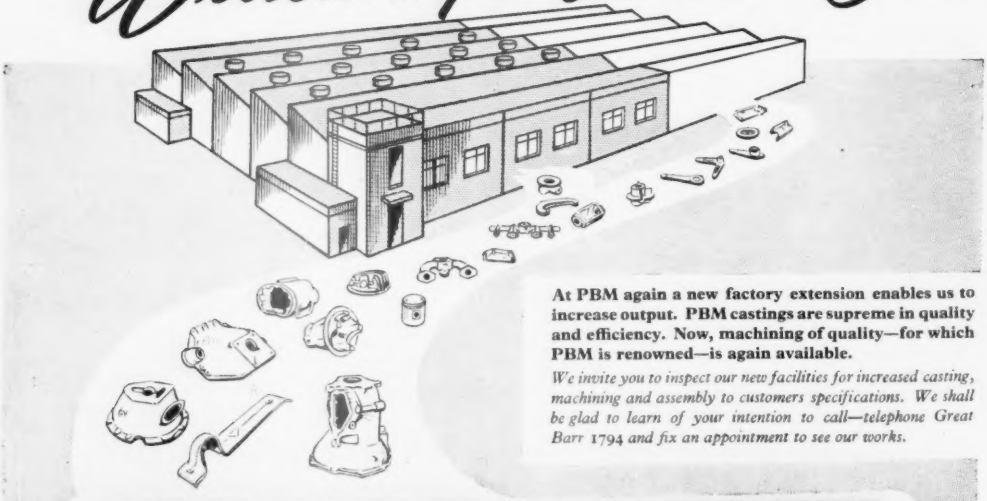

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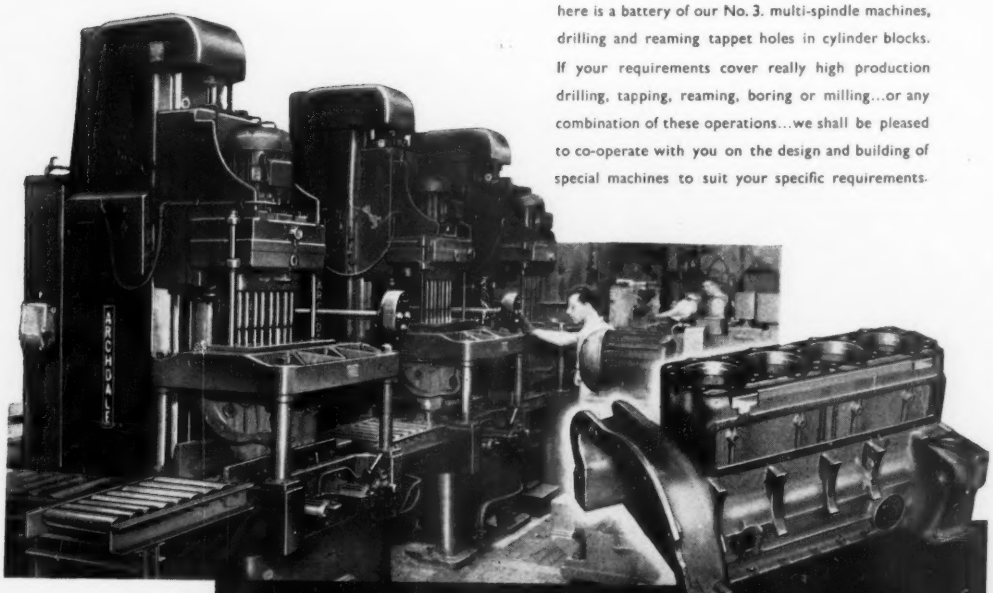

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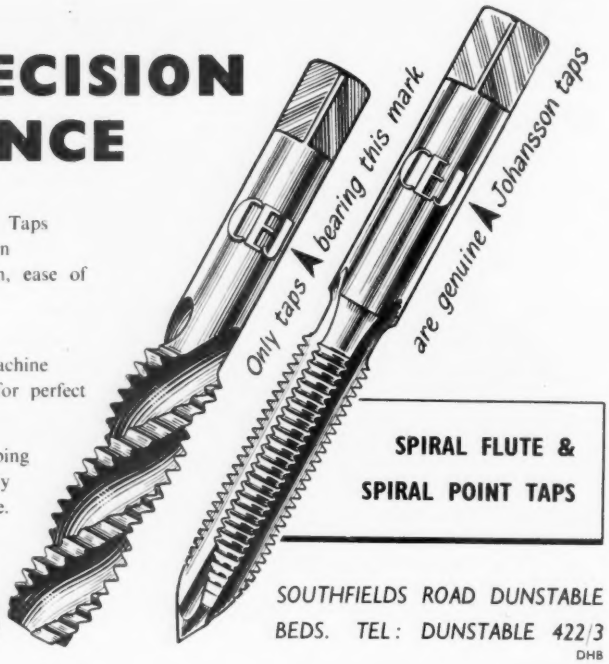

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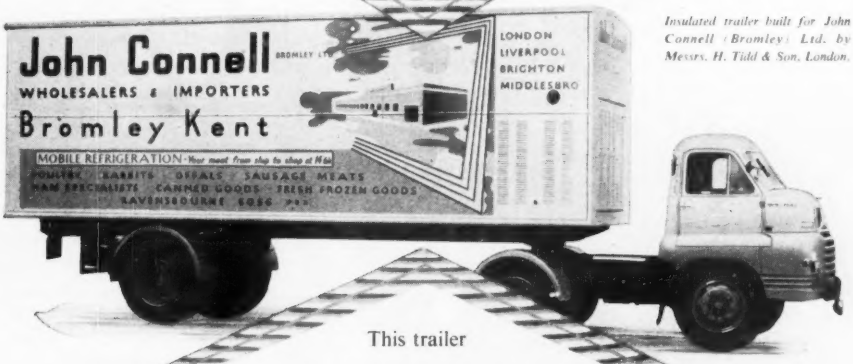

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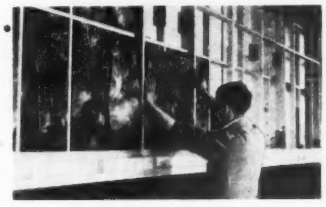

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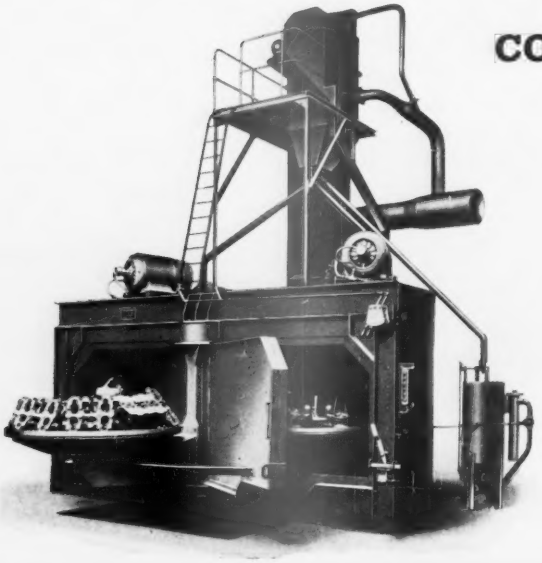

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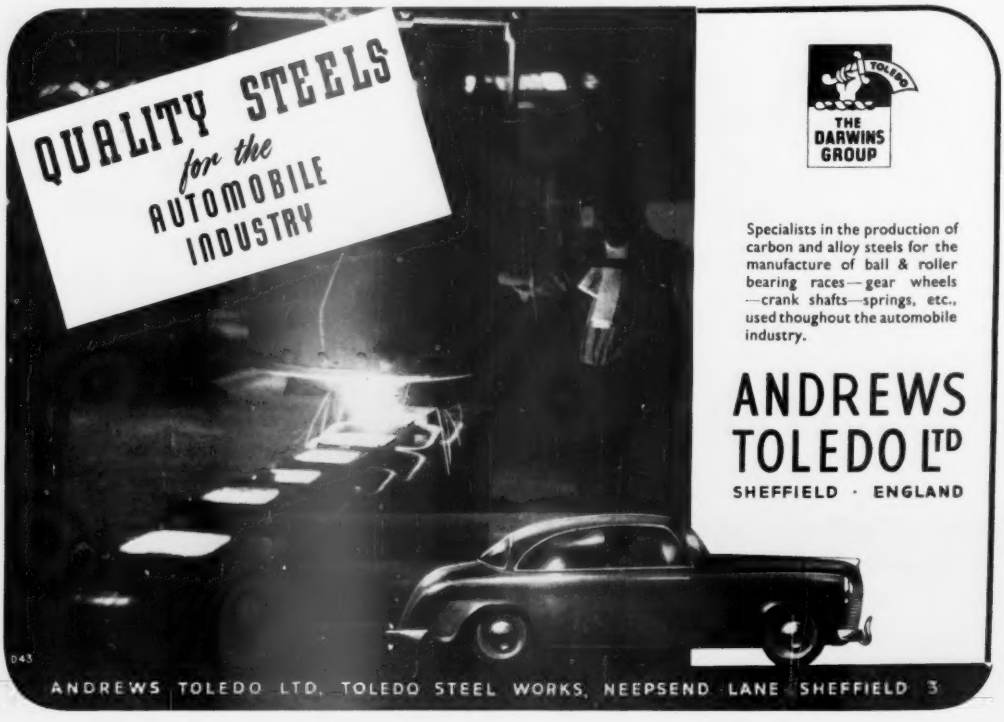
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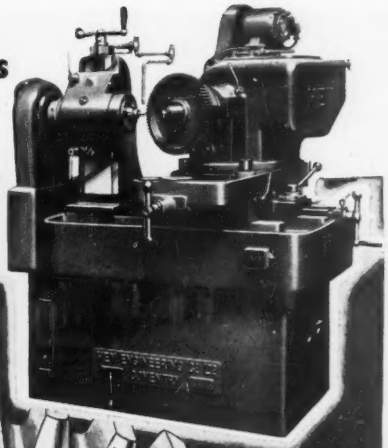

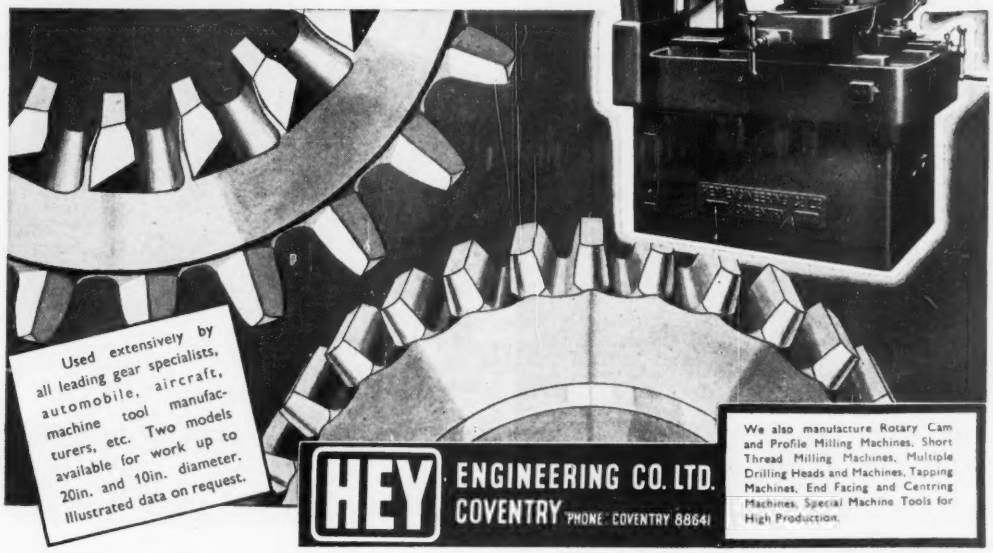

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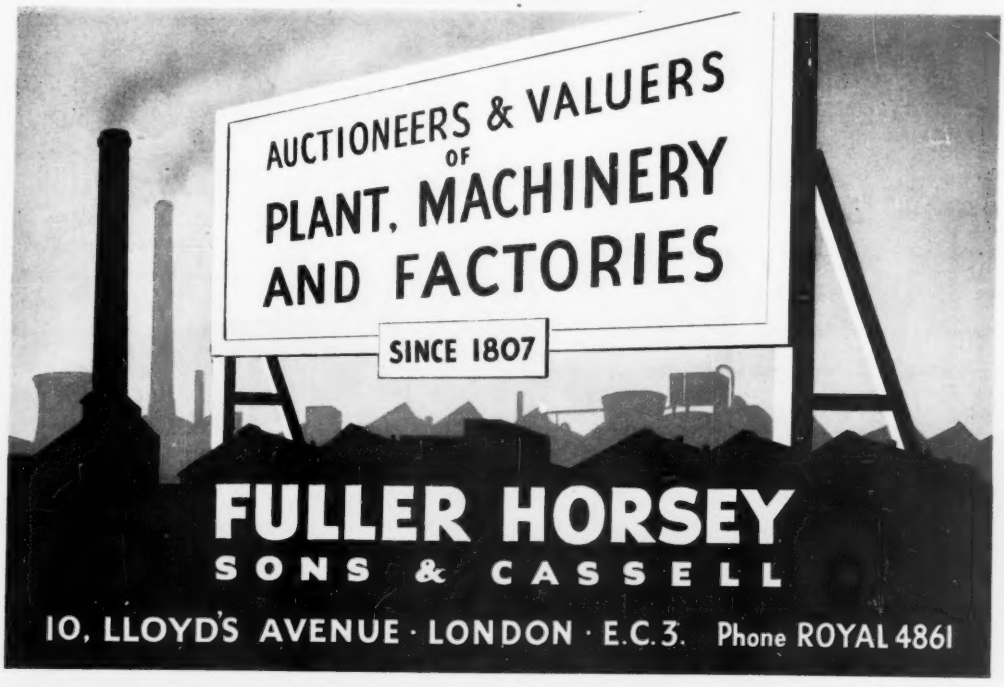

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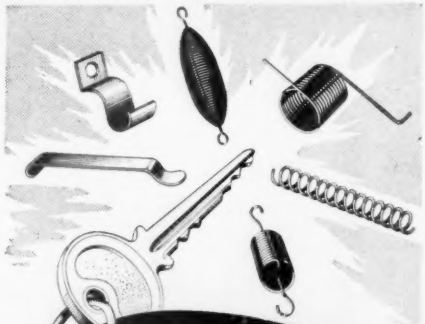

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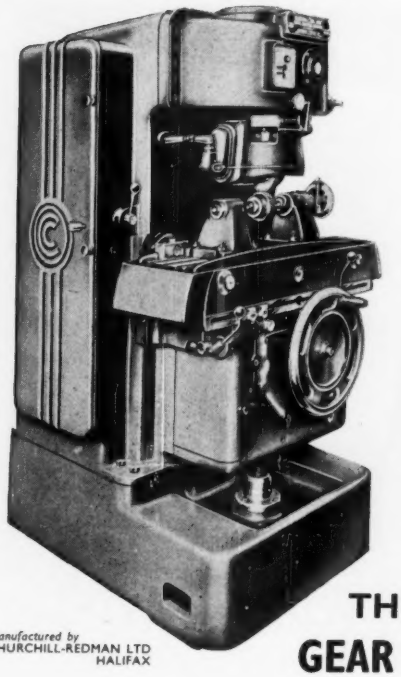

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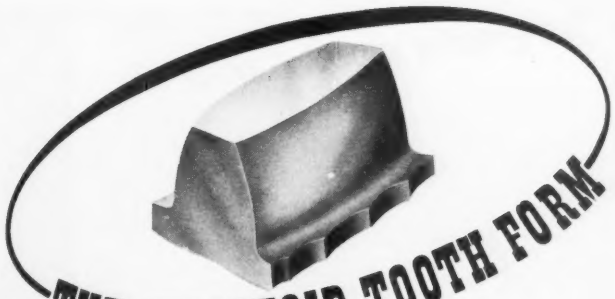

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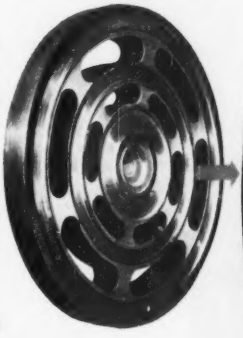

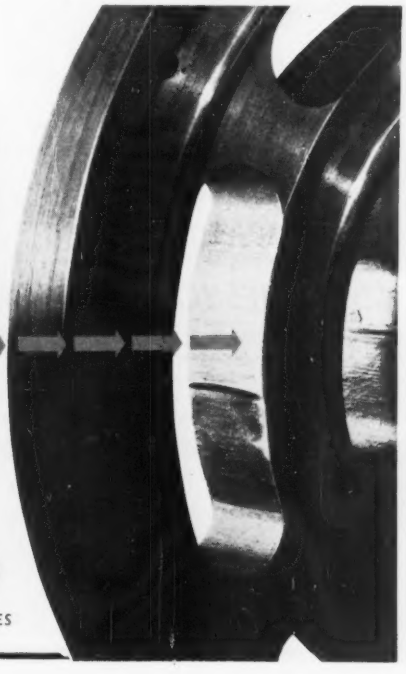

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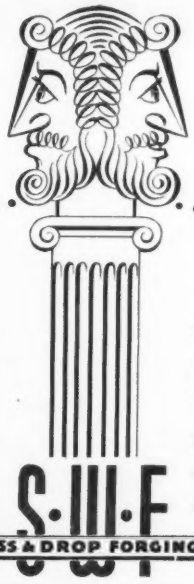

looking back...

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The Roman God Janus (the keeper of the door between the old year and the new, from whose name the word January originated) was depicted as having two faces—one to survey the past and the other to look ahead into the future.

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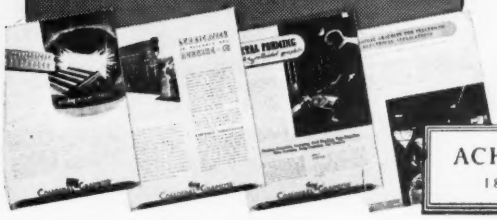

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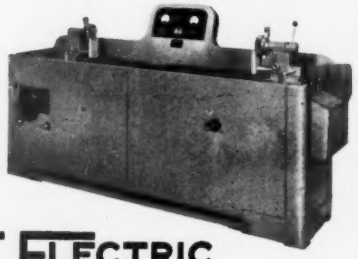

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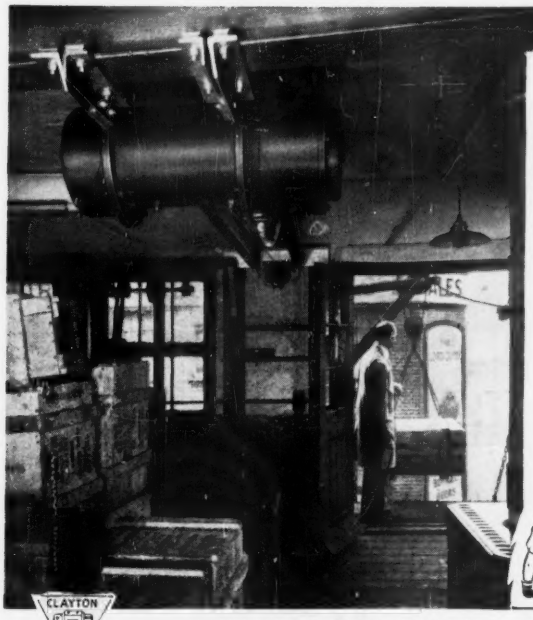

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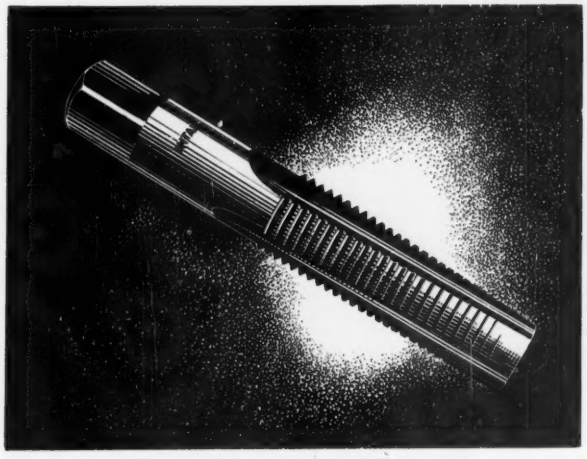

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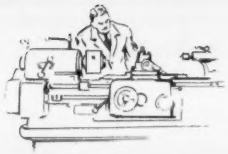

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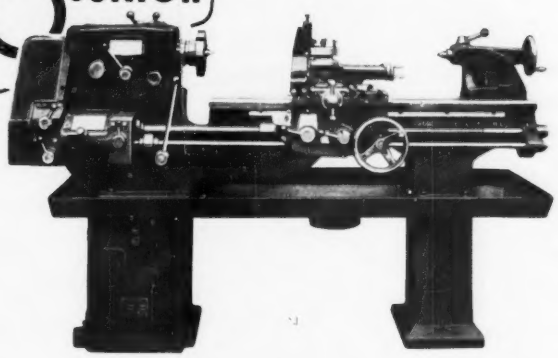

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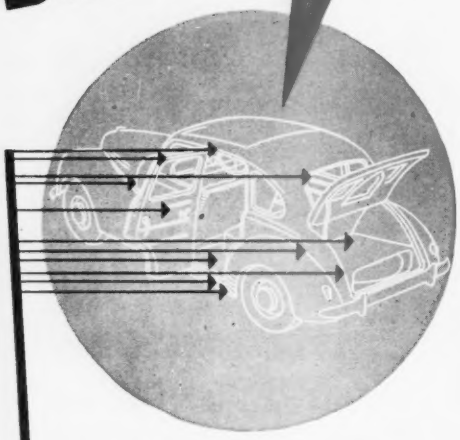

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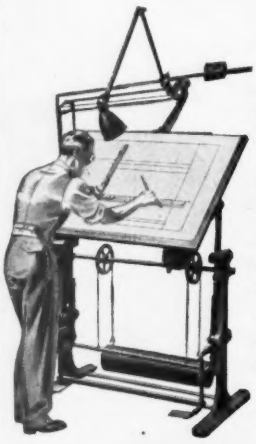

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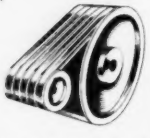

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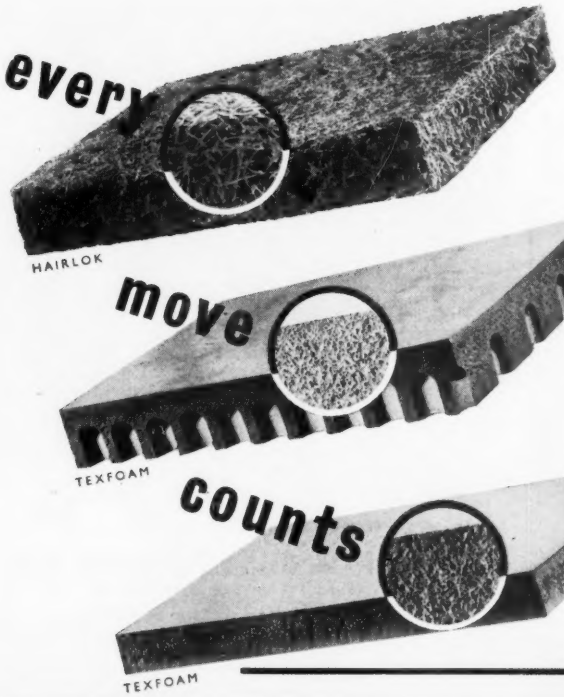

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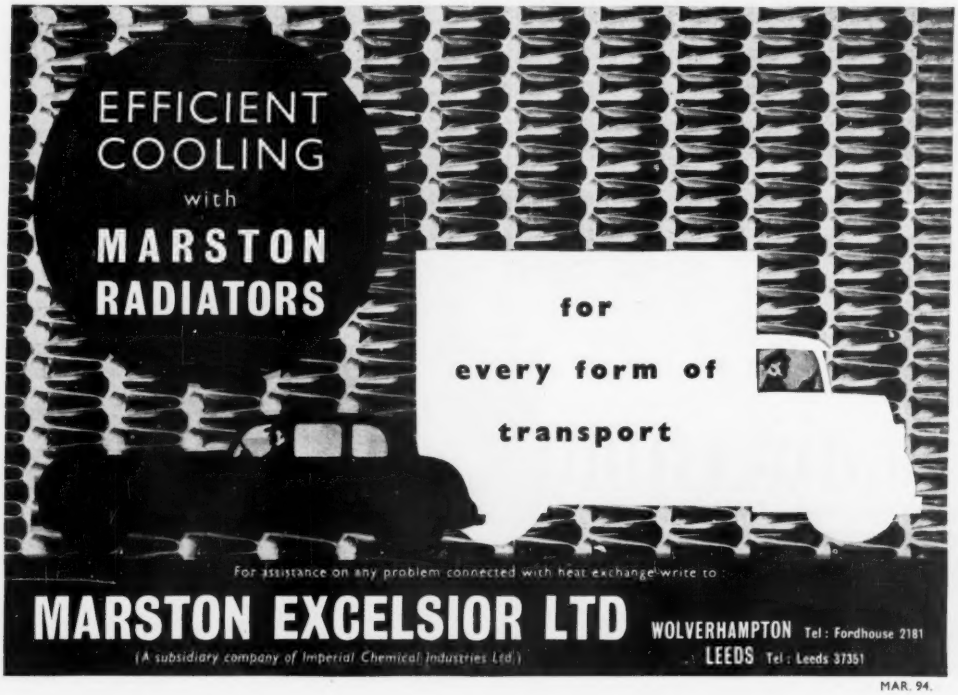

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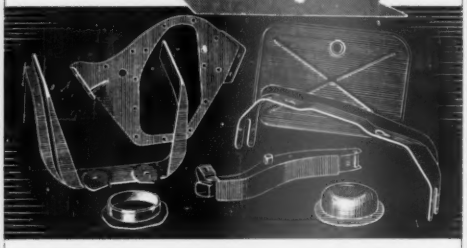

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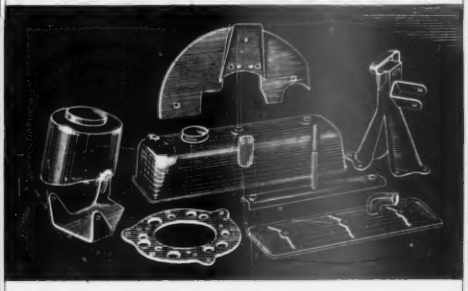

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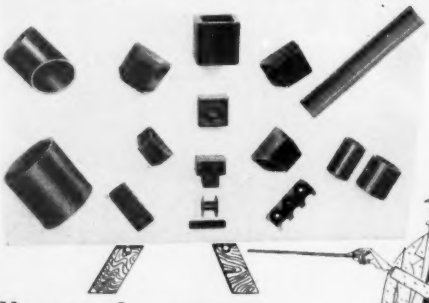

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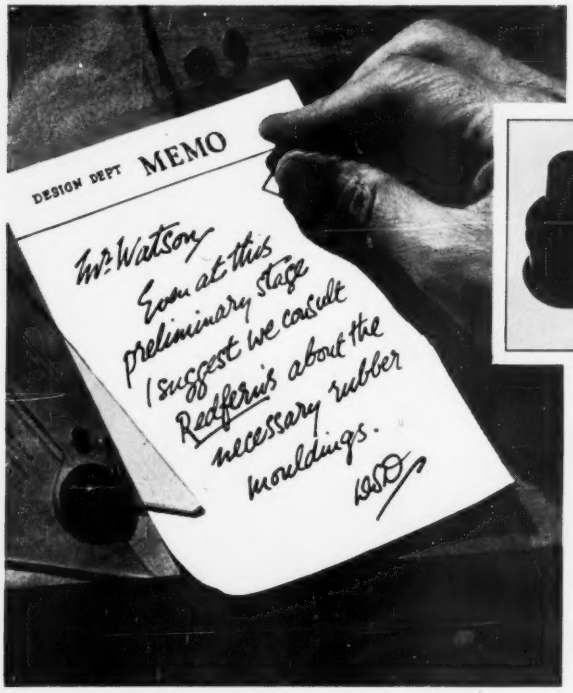
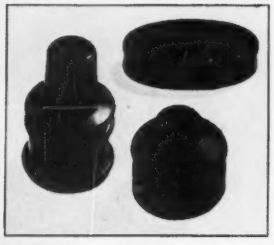


COMPONENTS



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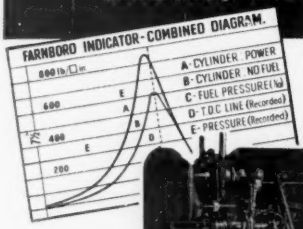

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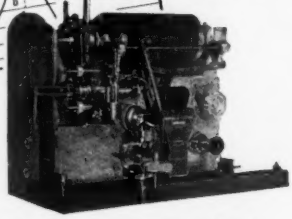

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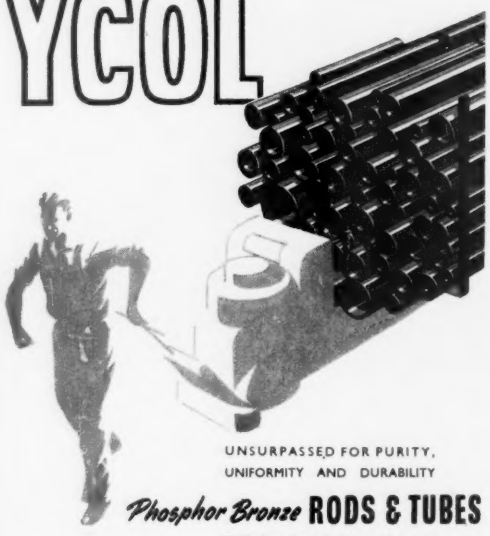

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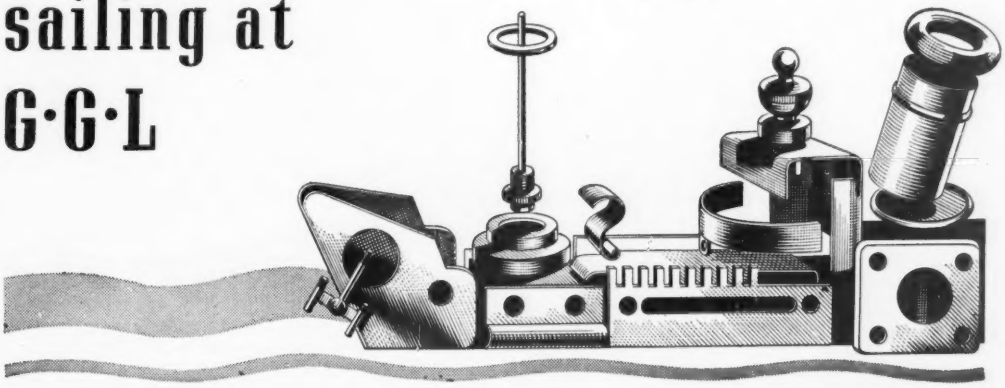

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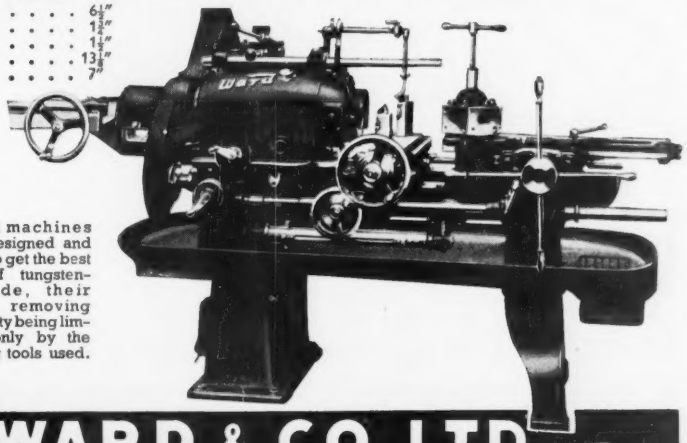

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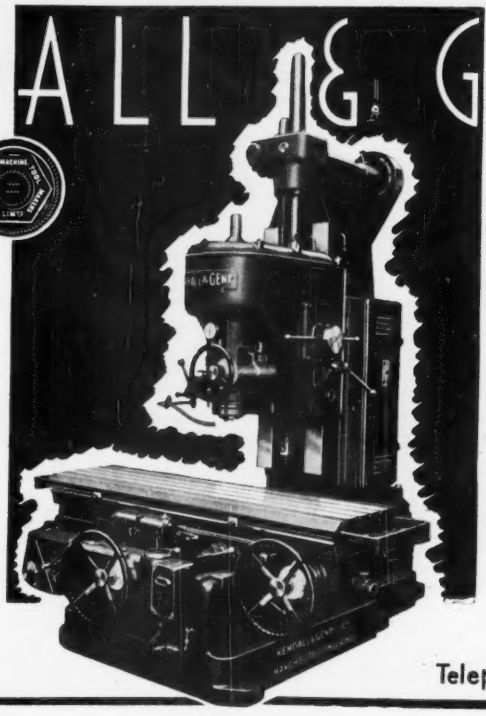

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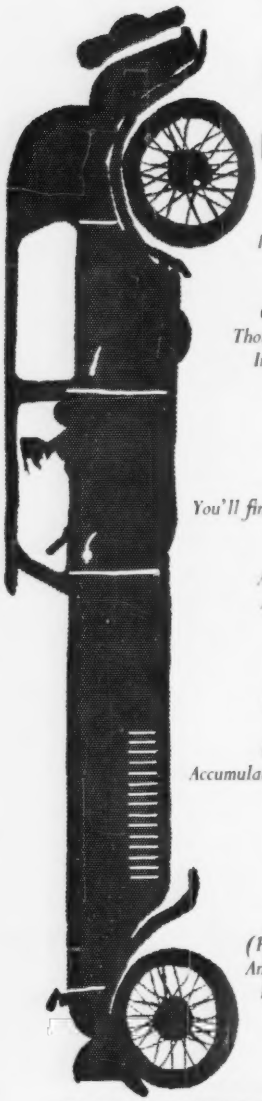

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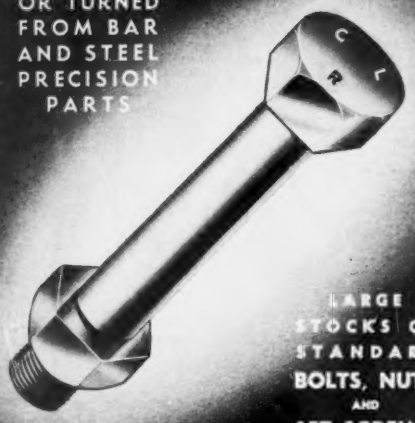

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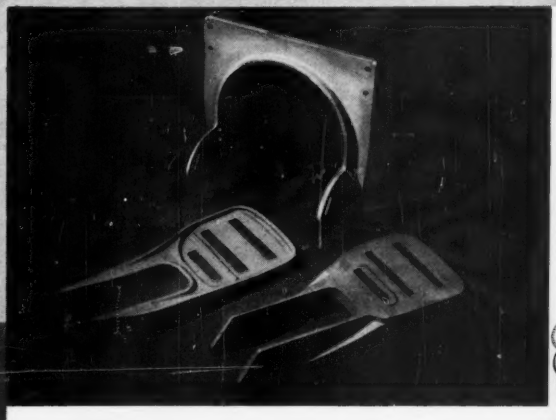

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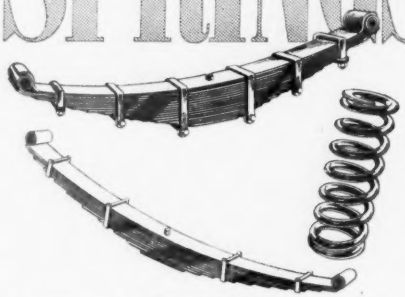

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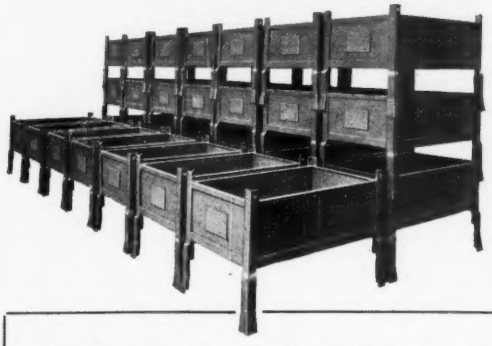

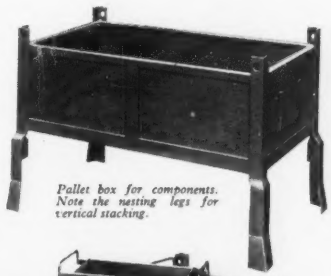

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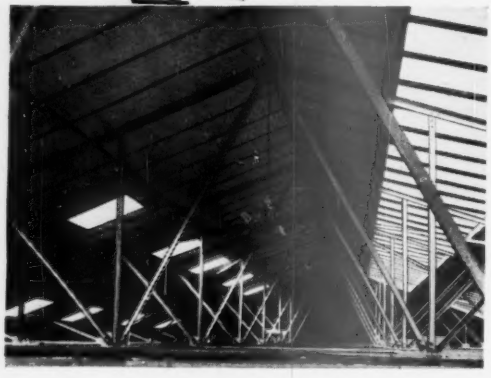

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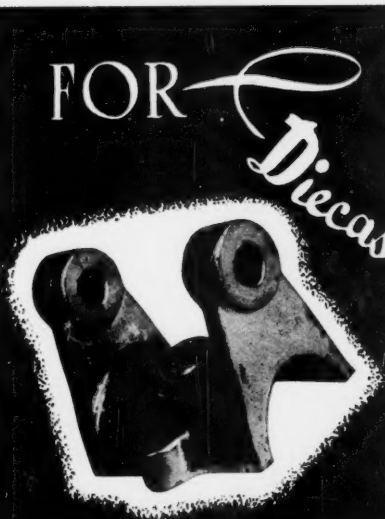

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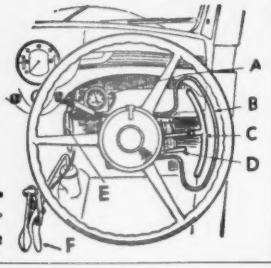

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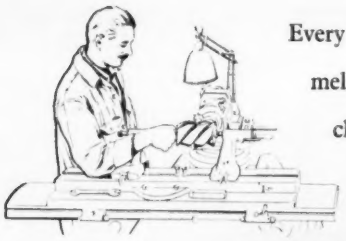

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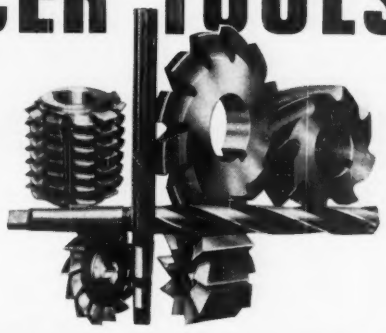

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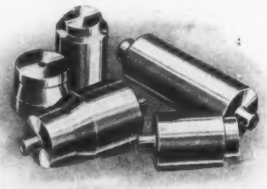

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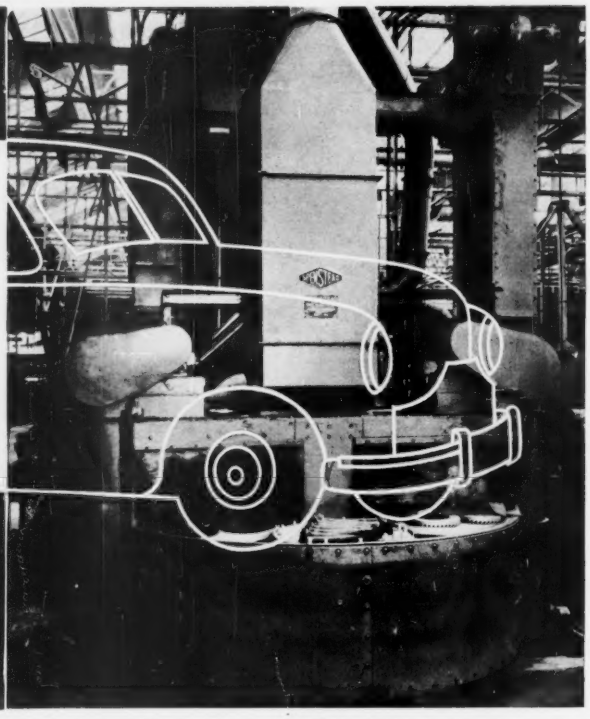

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